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THESIS FOR THE DEGREE OF PH.D.

THE MINE VENTILATING FAN.

By

James Norval Williamson, B.Sc.,

William Dawson Research Scholar,
University of Edinburgh.

Directed and supervised by Professor Henry Briggs, D.Sc., Ph.D.

Mining Department,

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INTRODUCTORY.

The orthodox colliery ventilating fan to-day is the centrifugal machine. While it appears the simplest installation at or in the mine, it is nevertheless one wherein great complexity exists regarding its efficient design; the general principles of the centrifugal ventilator are straightforward enough, but the particular details of its construction present numerous difficulties. Compared with the centrifugal pump dealing with water the design of a machine on similar principles to deal with air is a much more troublesome problem, when the marked difference in the physical properties of the two fluids is considered.

A study of the existing theory, design, and practice of the modern ventilating fan, supported by experimental investigation with a view towards possible improvement, particularly in design, would seem opportune for three principal reasons.

1. The economic exploitation of our richest national asset is becoming more and more a question demanding the utmost rigor in the elimination of all unnecessary wastage, both in labour and machinery. Those responsible for the successful working of our coal mines must therefore ensure that such a vital factor as ventilation is effected with the maximum all-round efficiency possible.

2. With the prospect of deeper and more extensive mining operations, and consequently higher underground temperatures, the problem of adequate mine ventilation assumes a more acute character.

3. In 1923, the Council of the Institute of Mining Engineers deemed it expedient to set up a Committee for the purpose of revising the existing theory of mine ventilation; the theory of the mine ventilator cannot be dissociated from such a revision.

Considering the first of the foregoing reasons, published /

published records[1] of fan efficiencies strongly suggest that there is much room for improvement in this connection. Indeed, it is highly questionable whether overall effic:iencies (i.e., the ratio of useful work done by the fan to the total energy supplied to the installation) over 50 per cent. are the rule rather than the exception, although the figures quoted in fan makers' catalogues would indicate otherwise. In all fairness to the fan makers, however, it must be admitted that the blame for the general low efficiencies of their products cannot be ascribed to them entirely. Modern centrifugal ventilators are designed to give their maximum efficiency over a fairly limited range of conditions and any variation from such limits results in a rapid fall in efficiency. The resistance to the flow of air through a mine usually varies considerably during its lifetime, and in the majority of cases, the fan operates for a comparatively brief period - if at all - under the conditions for which it was designed. Hence, when we keep in mind the fact that the fan is required to run continuously, 24 hours per day, throughout the life of the mine, the enormous financial loss involved in consequence of such low efficiencies is at once apparent.

That ineffective ventilation has a direct influence on the health, and therefore on the efficiency, of the miner, is too obvious a statement to require support. Generally, however, the inadequacy of the air circulation around the working places is more due to faulty distributive arrangements underground than due to the/

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1. For example, in the discussion of a paper, "Experiments on the Distribution of Air in Centrifugal Fans and on Re-Entry Phenomena", by Prof. Henry Briggs and the writer, (Trans. Inst. of Min. Engs., 1923-24, Vol. LXVII, pages 84-99), Mr. Samuel Hare, the then President of the North of England Inst. of Min. & Mech. Engs., stated: "I have come across cases where the efficiency was not higher than 25 per cent". (Discussion, page 244).

the ventilator itself.

Although it is strictly a future problem rather than a present one, the atmospheric conditions likely to be encountered in deep and extensive mines is engaging the attention of a large body of scientific workers. The Ninth Report of the Committee selected by the Safety in Mines Research Board to study this problem was recently issued [1]. It is not dry heat which matters but moist heat, and air in its passage through the mine absorbs considerable moisture, particularly at the working places. Experience and experiment have shown that the human body cannot stand a wet bulb temperature much beyond 80° Fahrenheit.[2] Both from the health and efficiency standpoints therefore, it becomes a vital necessity to maintain the atmospheric conditions of the mine within the physiological limits of the worker. The most practical solution of this future mining problem would seem to be a copious supply of fresh air kept briskly in circulation throughout the working places, rather than in the provision of an elaborate air-cooling plant as has been made in the Morro Velho Mine in Brazil [3]. The necessity for the installation and maintenance of an efficient ventilator, and of adequate methods of conducting the air around the mine becomes, then, a factor of primary importance.

While the third reason advanced for the opportuneness of a study of the ventilating fan is, from a practical point of view, somewhat overshadowed by the other two, it is nevertheless justified, particularly from the academic side. The theory of mine ventilation is /

1. "Control of Atmospheric Conditions in Hot and Deep Mines" (Ninth Report) Trans. Inst. Min. Engs., 1924-2 Vol. LXVIII, page 377.
2. "Modern Mining Practice" by R.A.S. Redmayne, Vol. IV, page 19.
3. "Air-Cooling Plant at the Morro Velho Mine, Brazil", by E. Davies, Trans. Inst. Min. Engs., Vol. LXIII, 1922-23, p. 326.

is at present in the melting pot, and principles, both simpler and more scienitific, will shortly be recommended by the Committee referred to. Since the classical expositions of Atkinson[1] and Murgue[2], very little advance had been made in this important subject until quite recently. In a paper[3] read before the Scottish Institute of Mining Engineers in 1921, Mr. D. Penman advocated the abandonment of Murgue's theories of the "equivalent orifice" and "orifice of passage" and a substitution of more direct methods of measuring the mine and fan resistances to the flow of air through them. While Penman cannot claim to have originated the idea of the direct method, he nevertheless made the most complete use of the electrical analogy regarding the flow of air through mines and fans which has yet been published, and was instrumental in arousing a discussion so healthy that it resulted in the Mining Institute taking up the matter authoritatively in the manner stated. The theory of the centrifugal ventilator is therefore involved in the work taken up by the Committee of that Institute.

The writer's investigations have been primarily concerned with the mode of action of the modern mine ven:
:tilator and its appendages; Part IV contains the record of this work together with conclusions arising therefrom. The evolution of the fan is discussed in Part I; the succeeding /

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1. "Theory of the Ventilation of Mines", by J. J. Atkinson; <u>Trans</u>. <u>N</u>. <u>E</u>. <u>Inst</u>. <u>Min</u>. <u>&</u> <u>Mech</u>. <u>Engs</u>. Vol. III, 1854-55;
2. "The Theory and Practice of the Centrifugal Ventilator", published in 1872.
3. "A New Method of Measuring Ventilation Resistances with Special Reference to the Operation of Fans in Combination", by D. Penman, <u>Trans</u>. <u>Inst</u>. <u>Min</u>. <u>Engs</u>. Vol. LXII, 1921-22, p. 34.

succeeding part is devoted to the theory of the machine, particularly from an efficiency basis; while in Part III a resumé of recent work connected with our subject is attempted.

At the outset, I should like to record my indebtedness to Professor Henry Briggs, under whose direction and supervision the experimental work herein described was carried out. He afforded me every facility in the performance of the work and was ever ready to discuss my difficulties; his personal interest, keen criticism and considered advice have been most invaluable. My thanks are also due to various members of the staffs of the Engineering Departments of Heriot-Watt College for practical assistance in many ways; I feel I cannot eulogise too strongly the spirit of willing co-operation which exists between the departments of this Institution.

PART I.

HISTORICAL REVIEW OF THE EVOLUTION OF THE MODERN MINE VENTILATING FAN.

HISTORICAL REVIEW OF THE EVOLUTION OF THE MODERN MINE VENTILATING FAN.

Furnace and Steam Jet Ventilators.-

The application of machinery to the purposes of ventilation of coal mines was slow in its early development. Mechanical ventilators were employed in metalliferous mines long before their introduction to coal mines was even considered. Indeed, in 1557, Agricola, in his well-known work, De re Metallica, gives particulars of various ventilating appliances, including a small hand-driven centrifugal fan; most of the ventilators he describes, however, are bellows actuated either by hand or by a horse-gin, and displacement machines driven by water power. Up to the end of the eighteenth century, coal mines were ventilated chiefly by natural means, supplemented by fire baskets suspended in the upcast shaft. Towards the close of that century, probably the first ventilating furnace was installed at North Biddick Colliery[1] in the north of England. This consisted of a chimney, 30 feet in height, erected on the surface, in which a furnace fire was kept burning to induce a flow of air towards it from the upcast shaft to which it was connected. Subsequently, when John Buddle, a prominent mining engineer of that period, had shown that there was as much likelihood of an explosion of gas occurring at the top as at the bottom of the upcast shaft,[2] the underground furnace ventilator became the recognised method of ventilating coal mines. What may be considered as the final stage in furnace ventilation was the provision of the "dumb drift"; this permits the furnace to be fed with intake air while the products of combustion pass along an inclined tunnel leading into the upcast shaft. This arrangement certainly helped to minimise the risk of firedamp /

1. "Historical Review of Coal Mining", page 130.
2. Ibid., page 133.

firedamp explosions as far as the furnace ventilator was concerned, but because of its potential danger, it was indeed surprising that its use was so long continued. The Coal Mines Act (1911), Section 31 (4) now prohibits its installation into all except small coal mines[1] in which the upcast shaft contains no inflammable material. Nevertheless, a few rêlics of this form of ventilator are still in evidence, an interesting example being that at Wallsall Wood Colliery,[2] a naked light mine, where it has been in continuous use since 1879. The capacity of this furnace ventilator is 100,000 cubic feet of air per minute at an average ventilating pressure of 7 lbs. per square foot while the average cost is about 1.16 pence per 100,000 cubic feet of air.

Contemporaneous with the furnace ventilation phase was the attempt to induce air currents through the mine by means of steam jets in the upcast shafts. Such efforts being encouraged by the natural cry for greater safety, many collieries adopted the steam jet system. One of the most notable examples of this form of ventilator was that installed at Seaton Delaval Colliery[3] in 1849, the plant consisting of 25 steam jets, each $\frac{5}{8}$th-inch. in diameter, the steam pressure employed being 35 lbs. per square inch, and the volume of air produced in the upcast shaft being 79,000 cubic feet per minute.. In 1852, a Committee of the House of Lords reported:-

> "Your Committee are of opinion -
> "That any system of ventilation depending
> "on complicate machinery is undesirable,
> "since under any disarrangement or fracture
> "of its parts the ventilation is stopped or
> "becomes inefficient.
> "That the two systems which alone can be
> "considered as rival powers are the <u>furnace</u>
> "and <u>steam jet</u>."

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1. Vide Coal Mines Act, 1911, Section 122, a "small mine" is one in which the total number of persons employed below ground does not exceed thirty.
2. "Historical Review of Coal Mining", Appendix, page 36.
3. Ibid., page 140.

"Your Commiteee are unanimously of
"opinion that the steam jet is the most
"powerful and at the same time least
"expensive method for the ventilation
"of mines".

However, shortly after this report was published a series of comparative tests were carried out to determine the relative efficiency of the furnace and steam jet forms of ventilators, and as a result, the latter fell into disfavour. Evidence of an attempt to revive the use of steam in the creation of ventilative currents is given in a recent British Patent Specification.[1] Therein appears a description of an invention for utilising the heat liberated by the condensation of exhaust steam for heating the air in the upcast shaft; flanged radiator pipes, placed in the path of the air, are used for conveying the steam. Although the inventor claims that such a radiator system is capable of maintaining the total circulation of air required for a mine, it is doubtful whether it can be considered further than as an auxiliary to the main ventilator in modern mines.

Early Mechanical Ventilators.-

Two factors which retarded the advent of mechanical appliances to the ventilation of coal mines in this country were (a) the report of the Committee of the House of Lords in 1852 (already referred to), and (b) the cheapness of the source of power in furnace ventilation. On the Continent, however, mechanical ventilators had met with so much success that, by the middle of the nineteenth century, they had almost entirely superseded furnace ventilation. The earlier types of ventilators were of variable capacity and were worked by the rectilinear movement of a bell or plunger. In /

1. British Patent Specification No. 212,802, 20/3/24. Fritz Heise, Bochum, Germany.
~~2. Mining Journal, Vol XIII, page 158.~~

In 1813, John Buddle described to the "Sunderland Society" the first mechanical ventilator introduced by him at Hebburn Colliery, six years earlier. This was simply an air pump of the exhausting type, consisting of a large wooden piston working inside a wood-lined chamber; it exhausted 6000 cubic feet of air per minute. One of the earliest pioneers of the mechanical ventilator was William Fourness of Leeds. In 1837, he invented a rotary air drum, which resembled a barrel churn. The Fourness fan[1] could deal with 4,500 cubic feet of air per minute at a cost of 1/- to 1/6d. per day. The power required to drive this fan was only 3 H.P. and it enjoyed a brief period of popularity. Probably the best examples of the air displacement machines which held the field in the earlier part of the nineteenth century are (a) the Hartz Ventilator, (b) Nixon's Ventilator, (c) Struve's Bell Ventilator, and (d) Root's Blower, descriptions and illustrations of which are found in the earlier text books.[2]

Early Screw-Propeller Fans.-

Many fans of the Archimedian screw-principle were patented, one of which was introduced at one of Earl Fitzwilliam's collieries by Benjamin Biram in 1842,[3] but was only partially successful, being influenced by the direction of the prevailing winds. Another class of ventilator which was unable to make much headway at that time was the screw-propeller type. Although there is little friction in the working parts of these /

1. Mining Journal, Vol. XIII, page 158.
2. Eg. See "Ventilation in Mines", by Robt. Wabner; trans: :lated from the German by Chas. Salter (1903) pages 143, 144 and 147, and Figures 77, 82, 83 and 88.
3 Mining Journal, Volume XII, page 397; Volume XX, page 57.

these fans, the internal resistance to the passage of air small, and their capacity large, factors all desired in

Figure 1.

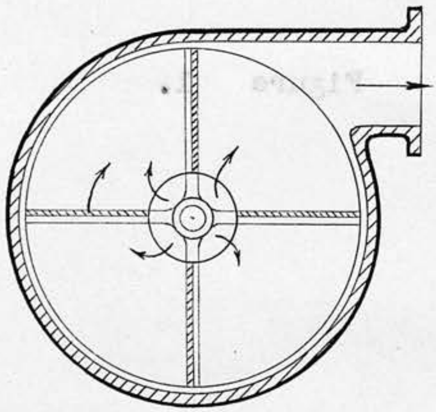

Figure I. Early Centrifugal Ventilator (17th Century).

these fans, the internal resistance to the passage of air small, and their capacity large, factors all desired in the mine ventilator, nevertheless, like the furnace, they are, generally speaking, unable to produce large ventilating pressures. Moreover, their efficiency is very low. However, recent researches in connection with air-screws by Mr. F. A. Steart,[1] indicate that a new era is about to be opened up for this type of ventilator. The subject of thd suitability of air-screws for mine ventilation is discussed in a later section.

The Development of the Centrifugal Type of Ventilator.-

The bulkiness, unreliability, limited capacity and low efficiency of the air displacement form of ventilator, led to its abandonment in favour of the centrifugal type. The oldest and simplest form of centrifugal fans is illustrated in Figure 1, Agricola having referred to its use in the Freiburg ore mines[2] early in the seventeenth century. It consists of four straight vanes carried on a horizontal shaft and enclosed in a circular casing with flat sides, the casing terminating in an effluent orifice. The fan was actuated by a hand crank. Air was drawn in through lateral apertures surrounding the shaft, and was discharged through the peripheral orifice by the centrifugal force generated by the rotation of the fan. The various forms of centrifugal ventilators now in use in mine ventilation have sprung from this simple hand fan.

The country which has played the principal part in the introduction of the centrifugal fan is Belgium. The reason /

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1. "The Application of Air-Screws to Mine Ventilation", by F. A. Steart, Trans. Inst. Min. Engs., Vol. LXVIII, 1924, pp. 310-322.
2. Vide Robert Wabner, "Ventilation in Mines", p. 151.

reason is not far to seek. The firedamp danger was more pronounced in Belgian mines than in those of any other country, and this, combined with the progressive development of deeper mining operations, made it imperative that better appliances be installed for the efficient ventilation of the mines there. Technical education was also on a high plane in Belgium at that period, so that, all things considered, the mining engineer of that country was perhaps pre-eminently suited for the development of the fundamental principles of the centrifugal fan and its practical application.

M. Letoret, formerly Principal of the Mons School of Mines, was the pioneer of the centrifugal fan for the ventilation of Coal mines. The first Letoret fan was installed at the St. Victoire Colliery, Agrappe.[1] Its design was practically identical with that illustrated in Figure 1 except for the casing of the latter; the Letoret fan was placed directly over the upcast shaft and between two vertical brick walls, an inlet being made in each wall. This simple ventilator was found to be very reliable and practicable, and it met with much favour. Its dimensions, and likewise its capacity, were very modest; its efficiency was low, but the power required was also very low, being only 3 to 4 H.P.

The Combes fan followed closely behind the Letoret machine; it was installed at the Grand Cornu Mine, Belgium. Combes realised that two ideal characteris:
:tics of such ventilators were (1) the reception of the air by the vanes without shock and (2) the discharge of the air without velocity. Neglect of these points in early designs was the principal cause of their very low efficiencies./

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1. Vide Robert Wabner, "Ventilation in Mines", page 153.

efficiencies. The profile of the Combes fan is given in Figure 2. The curvature of the blades was

Figure 2.

fans it was known that a considerable amount of re-entry

Figure 3.

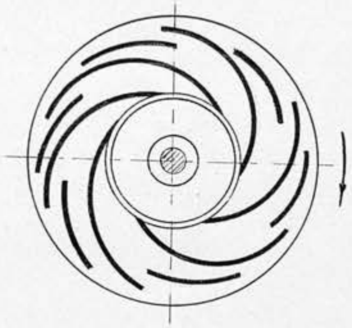

Figure 2. Combes Fan.

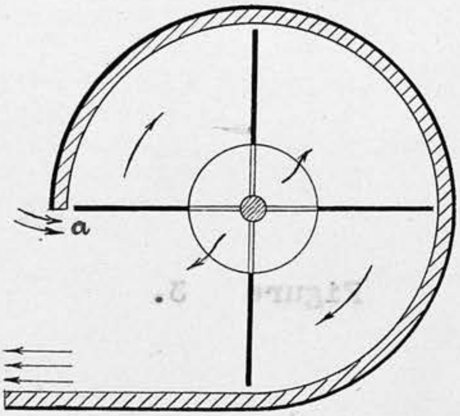

Figure 3. Fan used by Guibal in his Early Experiments (1855).

efficiencies. The profile of the Combes fan is given in Figure 2. The curvature of the blades was theoretically determined in order to embody the two characteristics mentioned, the tip of the blades being approximately tangential to the extreme circumference. In practice, however, these ideals were not realised, the efficiency being very poor and much below the anticipated figure.

The advent of the Guibal fan marked the most important advance in the design of these early ventilators; it was at once vastly superior to all its predecessors in simplicity, reliability and efficiency, so much so that we find Guibal fans running at many collieries to-day, both at home and abroad. To Guibal lies the credit of two innovations in design, (a) the casing, and (b) the expanding chimney, popularly termed the évasée. In the open running fans it was known that a considerable amount of re-entry of discharged air occurred between the vanes at the periphery. Immediately behind the tips of the vanes was a region of rarefaction, which induced a back flow of discharged air. Previous attempts to counteract this by introducing a larger number of vanes and giving them a backward trend at their tips, had proved of little avail. In 1855, Guibal and his pupil, Delsaux, made an experiment with a fan (see Figure 3) at the Esconffiaux Pit, Belgium, the fan being encased nearly all the way round. In the course of this test Guibal observed that air re-entered the casing at the point a; he therefore gradually reduced the sectional area of the discharge outlet until this re-entry ceased. He subsequently claimed that there was a definite relationship between the discharge outlet and /

and what he called the pit temperament;[1] thus, by means of a sliding damper, he adjusted the size of the effluent orifice relative to the pit temperament, so as to give the maximum efficiency. This sliding damper or shutter, is still one of the features of present day Guibal fans,[2] but its efficacy is of doubtful character. As one old writer expressed it, the Guibal shutter

> "seems to be mysterious in its action. This is
> "sometimes referred to almost as if it were some
> "source of power to help the fan when it got in
> "difficulties. As far as my experiences have
> "gone, it seemed to act the part of scapegoat to
> "all the shortcomings of the Guibal; it always
> "seemed unfortunately to be in the wrong place,
> "and, therefore, a fan in which it was dispensed
> "with would probably be desirable"[3].

Modern installations do not include the original Guibal casing, as shown in Figure 4, but rather favour the volute or spiral casing, Figure 5. Obviously, a fan will run more smoothly when discharging all round its circumference than when discharging in a series of intermittent spirts. Moreover, the spiral casing fulfils more or less the real function of the close casing; its gradually increasing cross-sectional area affords space for the momentary reception of the correspondingly increasing volume as it is being discharged from A to B (Figure 5).

Combes' unsuccessful effort to solve the problem of /

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1. The pit temperament was expressed by the ratio $\frac{Q^2}{h}$ where Q is the volume of air circulating and h, the ventilating pressure.
2. The Walker "Indestructible" Fan, which is simply a modernised Guibal provided with a commercial name, also possesses this adjustable shutter.
3. "An Account of a New Ventilating Fan", by T. J. Boulker. Trans. N. E. Inst. of Mech.& Min. Engs., Vol. XXXI, 1881-1882. Mr. Boulker, in the discussion of his paper.

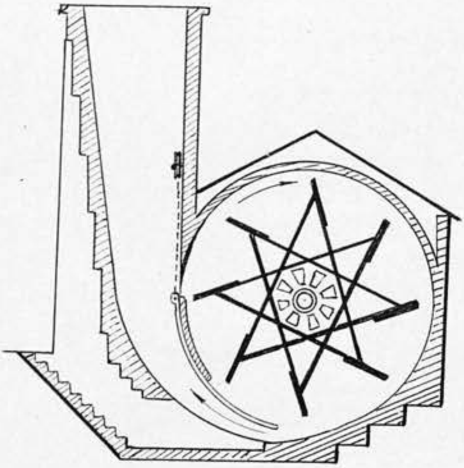

Figure 4. The GuibalFan.

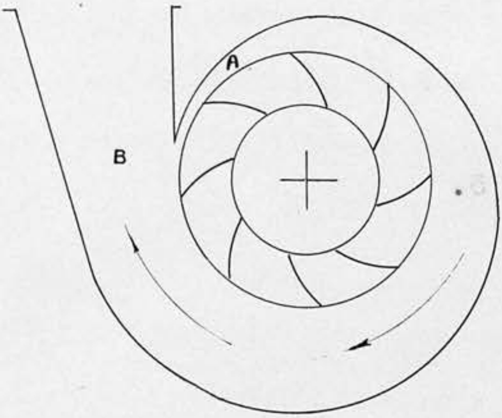

Figure 5. A Fan working in a Spiral Casing.

of the high velocity air discharged from the fan has been already referred to. Guibal, however, discovered a much better means of converting the velocity energy contained in the air discharged from the fan into pressure energy; he surmounted the enclosed fan with what we now call the évasée. Thus, the air coming from the fan passed through a gradually increasing area which consequently caused a reduction in its velocity, and, in accordance with the well-known Bernoullian theorem, the resulting loss of kinetic energy was balanced by a corresponding gain of pressure energy. The function and correct design of the évasée forms the subject of a later section.

The dimensions of the Guibal fan were large; machines up to 50 feet in diameter, 12 feet wide, and provided with ten vanes have been built. They are capable of producing moderate depressions only; when higher depressions are desired and an increased volume, the diameter of the Guibal would require to be increased in order to give the necessary peripheral velocity. The dimensions of this fan are already excessive however.

One of the first centrifugal machines introduced into the mines of this country was installed at a colliery near Paisley in 1827;[1] it resembled the winnowing machines used by farmers at that time for cleaning corn. It was placed horizontally within a circular casing over the mouth of the upcast shaft. Towards the middle of the century, a centrifugal fan was designed by William Brunton and installed at Gelly Gaev Colliery, Glamorganshire.[2]/

1. "Historical Review of Coal Mining", page 141.
2. Ibid, " 144.

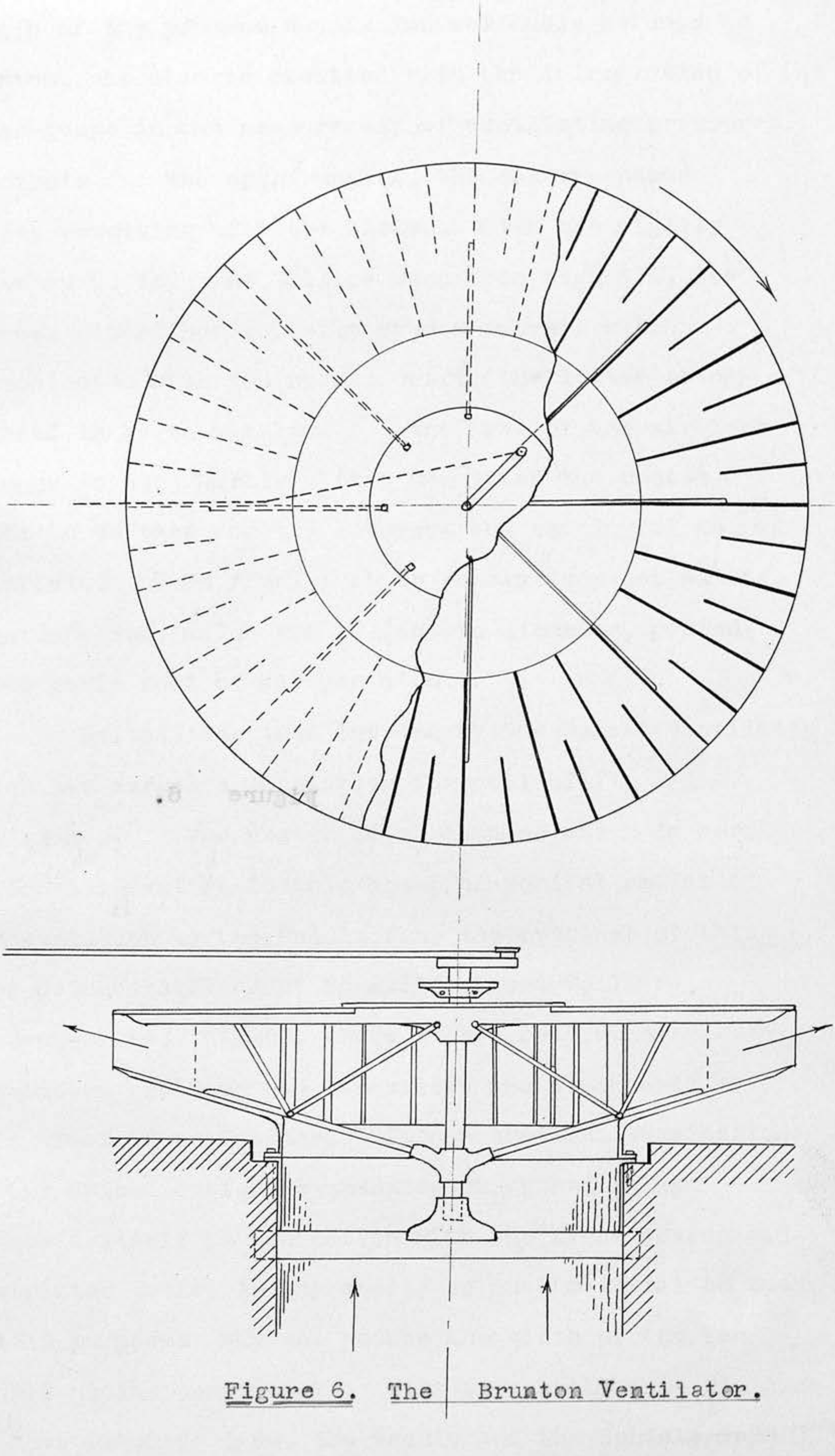

Figure 6. The Brunton Ventilator.

Glamorganshire. Brunton, who was trained in the Soho Works of Messrs Boulton and Watt, was a very able engineer. Figure 6 illustrates his ventilator. The origin of the present Waddle Fan obviously belongs to Brunton, who also is credited with the introduction of the water-gauge in the measurement of ventilating pressures. Brunton's fan was open running, the saucer-shaped casing revolving with the blades, which are rigidly attached to it. As will be seen from Figure 6, the fan was placed horizontally over a culvert which communicated with the upcast shaft, the latter being covered in by an air-lock. The idea of the air-lock belongs to John Martin; this permitted the upcast shaft to be used for the lowering and raising of men or materials. When running at 95 revolutions per minute, Brunton's fan, which was 22-feet in diameter, passed 18,000 cubic feet of air per minute.

Britain was then invaded by the Guibal ventilator which had earned a reputation for reliability and efficiency. The next British machine which is worthy of special mention in this brief historical review of fan evolution is the Waddle fan, the original of this type being installed at Bonsville Court Colliery, Pembrokeshire, in 1863, where it has been running ever since, convincing proof of the durability and dependability of this class of ventilator. Upon a critical examination of the Guibal design, a question which naturally suggests itself in connection with the close casing and restricted outlet is why employ so much material to such little purpose; why not reduce the width of the fan itself at its periphery? This is exactly what was done in some later designs, the Waddle and the Schiele fans being perhaps the best of the earlier types. The original Waddle idea aimed at making the passage from the /

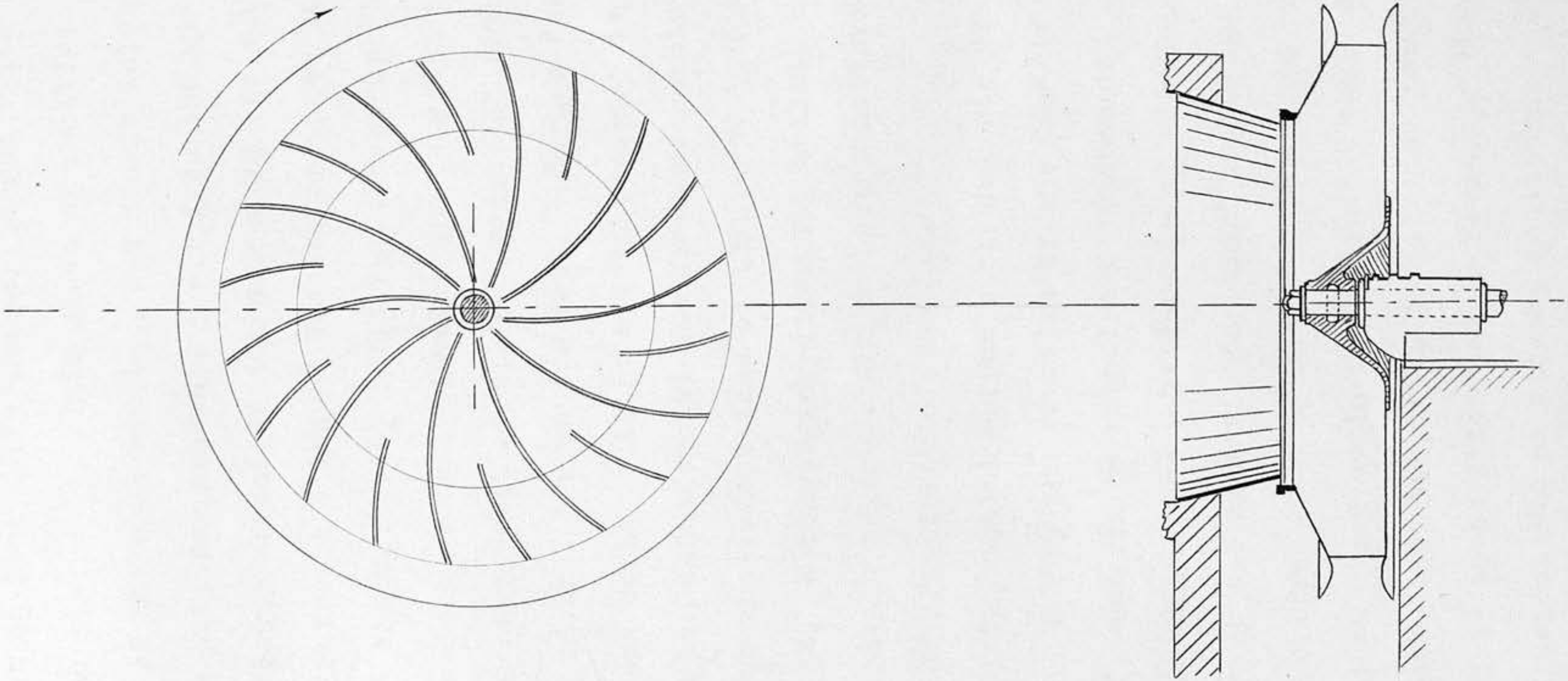

Figure. 7. A 24-foot Diameter Waddle Fan.

the throat to the periphery of uniform area. By such an arrangement, the centrifugal forces became much more equally balanced than could be expected in the original Guibal with its limited peripheral discharge. The dimensions of an old type Waddle may be 30-feet in diameter with 12-feet diameter inlet, and the width at the throat $3\frac{1}{2}$-feet while the peripheral width is not more than $1\frac{1}{4}$-feet. This class of ventilator is essentially of the open-running, and single inlet type, and intended only for exhausting air from the mine. A modified évasée effect is introduced at the outlet. Figure 7 illustrates a modern Waddle fan which is installed at the Wellesley Colliery, Fifeshire; this fan is 24-feet in overall diameter and is rated to give 210,000 cubic feet of air per minute at a ventilating pressure of 26 lbs. per square foot. Larger Waddle fans have been built, but the modern tendency is to reduce the size and increase the speed of revolution. The outlet of the latest fans of this type is shown in Figure 63, a converging-diverging outlet, a feature which the makers claim to have considerably augmented the efficiency of the fan.

In 1872, Daniel Murgue published his classical work on "The Theory and Practice of the Centrifugal Ventilator", wherein he enunciated fundamental principles on which the relationship of the fan and mine depend, making use of a fiction which he called the "equivalent orifice", and by means of which the relative resistances of mines could be indirectly compared. Murgue's methods of comparing ventilation resistances are now likely to be discarded in favour of those more direct and scientific as already mentioned. Nevertheless, Murgue's work had a profound influence on the theory of the subject, and led to general improvements in the construction /

construction of the centrifugal ventilator. Keen competition existed in the fan making industry, the Guibal being soon followed by the Waddle (already discussed) Schiele, Walker and Capell fans. All these ventilators have passed through evolutionary stages of their own and are numbered among present day ventilators. They also belong to the large class of fans and have comparatively few vanes, the curvature of which is backward[1]. Murgue made out a strong theoretical case for vanes having their convex sides reversed, and radial at their tips as in Figure 8. A prominent English mining engineer of that period was so fascinated with Murgue's reasoning that he had a fan built according to his (Murgue's) ideas, and publicly announced when the "machine perfect in every respect" was to be set in operation. The fan proved a failure.

In 1892, a Belgian commission[2] carried out an exhaustive series of experiments on mine fans; in all, 16 fans were tested, 4 each of the Guibal, Ser, Capell and Rateau types. The object of this Commission was to compare these ventilators from a practical basis, attention being devoted to (1) quantity discharged, (2) overall efficiency, (3) reliability, (4) cost of installation, and (5) advantages and disadvantages of ac: cessories. However, due to lack of standardisation in the methods of test and to incorrect measurement of the ventilating pressure, the results of these tests are of little value.

Seven years later, a joint committee of the North /

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1. The latest form of Capell, however, has vanes with a curvature in the direction of rotation.
2. "Les Ventilateurs de Mines", Revue Universelle des Mines, Vol. XX, 1892.

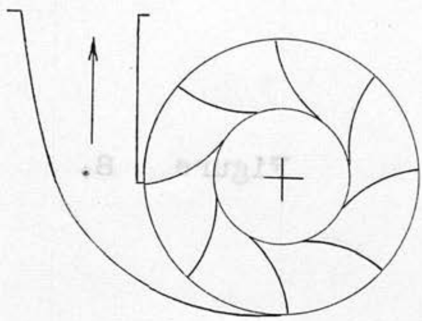

Figure 8. Profile of Vanes suggested by Murgue.

North of England Institute of Mining and Mechanical Engineers, and the Midland Institute of Mining, Civil and Mechanical Engineers, also carried out a series of tests on ventilators in an effort to arrive at some conclusion regarding the best type of fan suitable for mining purposes.[1] They tested four each of the Guibal, Schiele and Waddle class of centrifugal fans, collieries being selected where two fans of different types were installed capable of working either singly or together on the same air shaft. Unfortunately their otherwise careful work was also rendered practically useless because of the wrong method adopted in obtaining the value of the depression - the end of the water-gauge limb inside the fan drift was pointed towards the fan, and even although it was coated with a sleeve of flannel the depression recorded would be of too high a value. However, the results obtained by this Committee placed the Guibal fan first, with "a maximum percentage of useful effect of ventilator and engine of 54.8 per cent"; the Schiele and Waddle efficiencies were somewhat similar and slightly below that of the Guibal. While the main object of the Committee was not realised, much useful information on the subject of fan efficiencies was collated.

With the introduction in mining of the relatively high-speed electric motor, it was evident that smaller diameter, and therefore quicker running fans capable of developing higher depressions would have to be adopted in order to take the fullest advantage of the electric drive. The forerunner of this class of ventilator was the Sirocco, invented by the late Sir Samuel C. Davidson, and it was soon followed by /

1. The report of this Committee is published in Vol. XVII, Trans. Inst. Min. Engs., 1898-1899, (pages 482-583).

by others of somewhat similar design, such as the Keith, Barclay and Jeffrey fans. Some important new features were embodied in the design of this latest class of ventilator, among which are:-

(a) fans were made smaller in diameter and relatively wider than formerly - they were of drum shape;

(b) the fan-inlet was nearly equal to the diameter of the fan itself;

(c) vanes were made of shallow depth, much more numerous, and the curvature was forward (i.e.? in the direction of rotation).

(d) relatively higher speeds and depressions were obtainable.

The larger number of vanes tended to minimise re-entry, and their forward curvature was responsible for the air being discharged from the fan periphery at a resultant velocity considerably greater than that of the fan runner itself, a feature which accounts for the high volumetric yield of such fans. However, the makers of most of the older types of fans have endeavoured, with greater or less success, to bring their particular machines up-to-date, by reducing the diameter and increasing the speed of revolution. Nevertheless, during the past two decades, makers in general do not seem to have attempted any further improvements in the design of ventilators, a state of affairs which may be largely accounted for by the absence of any well-directed criticism.

PART II.

THE CENTRIFUGAL VENTILATOR IN THEORY AND PRACTICE.

SYMBOLS USED IN PART II.

U = the radial velocity of air discharged from fan-blade tips.
U_1 = the radial velocity of air at instant of entering blades.
u = the radial velocity of air at any point P, distant r from fan centre.
V = the peripheral velocity of fan = tangential velocity of air.
V_1 = the tangential velocity of air at inner edge of fan blades.
v = the tangential velocity of air at any point P.
V_b = the velocity of air relative to the blades.
V_r = the resultant velocity of air emergent from fan blades.
V_d = the mean velocity of discharged air.
H = the theoretical depression in feet of air-column.
H_A = the actual or effective depression in feet of air-column.
h = the effective depression in inches of water-column.
h_o = the depression due to the resistance of the fan - in inches of water-column.
G = the theoretical depression in inches of water-column.
Q = the quantity of air circulated in cubic feet per minute.
q = the quantity of air circulated in cubic feet per second.
b = the width of the blades at the periphery, in feet.
R = the radius of the fan to the blade-tips, in feet.
r = the radius of any point P.
(R = the resistance of the mine in Atkinsons)
or ( ) Equation (23)
(r = the resistance of the fan in Atkinsons)
w = the weight of a cubic foot of air.
w_1 = the weight of a cubic foot of water.
= the angular velocity, in radians per second.
M = the manometrical efficiency of the fan.
α = the angle of blade-tips as measured by the tangent to them and the normal.
β = the angle which the inner edge of the blade makes with the radius.
θ = the angle of blade-tips at any point P (measured as α).
g = the gravitational constant.
N = the number of revolutions of fan per minute.
l = symbol used in analysis of energy losses in fan.
k = a constant.

THE THEORY OF THE CENTRIFUGAL VENTILATOR.[1]

INTRODUCTORY.

This subject has been one of the most controversial in the science of mining, and one upon which a plethora of mathematics has been erected from, in many cases, false premises. The latter fact has, in consequence, done much to retard progress in the development of a more perfect theory, which would at once be both simple and scientific for all concerned, and materially help the management of our collieries in readily understanding the vagaries of action when the conditions under which the mine ventilator was operating were apparently but slightly altered. In the present treatment of the subject, the typical British Colliery ventilator, i.e., the exhausting fan, is considered, although with but slight modifications the arguments put forward will hold equally well for the compressive ventilator.

The Principle and Purpose of the Centrifugal Ventilator.-

In its general form a fan consists of a number of blades or vanes, which are rotated in a casing. Air enters the centre or inlet of the fan and is discharged through the blades round the periphery. The action of the blades produces a centrifugal force, which is proportional to the square of the speed of rotation of the fan; this force causes the air to move outwards from the inlet to the periphery. Work is thus done upon the air by the fan, the effect of which is to increase the velocity of the /

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1. In the preparation of several sections of Part II, I have received considerable guidance from the class lectures on the subject delivered by Professor Briggs.

the air as it leaves the tips of the blades, and also to increase its supply of pressure energy. In consequence of its velocity of emergence, the air possesses a considerable store of kinetic energy which is recovered to a more or less extent by some form of volute casing and by an expanding chimney or évasée.

In front of the vanes, then, the air is at a higher pressure than within the fan. Indeed, if we consider the mine, the fan drift, the fan and its adjutages, and the external atmosphere as a closed circuit, the region of lowest pressure in this circuit will be the fan centre. Air is thus made to flow round this circuit by the difference in pressure created by the fan.

It therefore follows that an exhausting fan is installed at a mine for the express purpose of producing a partial vacuum. As several writers have pointed out, the mine air circuit forms a close parallel with the electrical circuit. The position of the battery or dynamo in the latter circuit is analogous to that of the fan in the ventilating circuit of the mine with this difference - the fan induces a negative pressure whereas the battery or dynamo creates a positive pressure. Further, the volume of air flowing through the mine varies directly as the square root of the negative potential[1] created by the fan, and inversely as the mine resistance, which relationship, except for the index of the pressure, concurs with that existing between the current, potential and resistance of the electrical circuit, vide Ohm's Law.

Since /

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1. This is not strictly true; the correct relationship between volume and pressure is a matter which the Committee on Ventilation is to investigate.

Since air is induced to flow through the fan, its design must be such as to offer as little resistance as possible to the fluid in its passage from the fan-drift to the external atmosphere. Eddying and frictional losses can only be minimised by careful design, particularly as to the reception of the air by the blades and its ultimate discharge from them.

In thermodynamics the performance of a heat engine is compared with one working on the Carnot Cycle or the Rankine-Claudius' Cycle. Similarly, in this discussion of the ventilator, it would seem desirable to adopt an aero-dynamically perfect ventilator as a standard for comparison and a means of assessing the efficiency of the actual machine.

The Perfect Ventilating Fan.-

It may be taken as axiomatic, that an aerodynamically perfect ventilator, if such were possible to construct, would discharge air uniformly at all points of its peripheral area. Under these ideal conditions, the radial component of the effluent velocity, U, will also be uniform at all parts of the periphery. Hence, neglecting the thickness of the blades,

$$U = \frac{q}{2\pi Rb} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where U = the radial velocity of emergence at the tips of the blades, in feet per second.

q = volume of air delivered in cubic feet per second.

R = radius to the peripheral edge of the blades in feet.

and b = width of blades at the periphery in feet.

(a) Influence of the Shape of the Blades at their Peripheral Extremity.-

Before the theoretical depression created by our hypothetically perfect fan can be determined, we must /

must first consider the influence of the shape of the blades at their peripheral extremity on the velocity of the effluent air. In most mining text-books published prior to 1924, the theoretical depression created by a fan is stated to be given by the formula:-

$$H = \frac{V^2}{g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where H = theoretical depression measured in feet of air column.

V = peripheral velocity of the fan, in feet per second.

and g = the gravitational constant.

This formula is given in a general way and the student is left to infer that its application is likewise general. Murgue, in a supplement[1] to the treatise already referred to certainly deduced the above formula, but at the same time qualified it as being applicable only to an "ideal ventilator". (Murgue was obscessed with the idea that the blades should be normal to the circumference at their tips, i.e., radial, and his "ideal ventilator" was thus of such design - see Figure 8). This, however, affords no excuse for the numerous writers who adopted such formula as being of general utility.

Figures 9 and 10 illustrate portions of two fans, one with backward-trending blades at the periphery and the other with its blade tips curved forward in the direction of their rotation; both types of blade curvature are in common use. The angle, α, made by a tangent to the blade at its tip and the normal is the most convenient way of assessing the inclination of the blades at their outer extremities; when α is given the positive sign it indicates that the curvature is /

1. "The Theories and Practice of Centrifugal Ventilating Machines", by D. Murgue (1883) page 19. Translated by A. L. Steavenson. (E. & F. N. Spon).

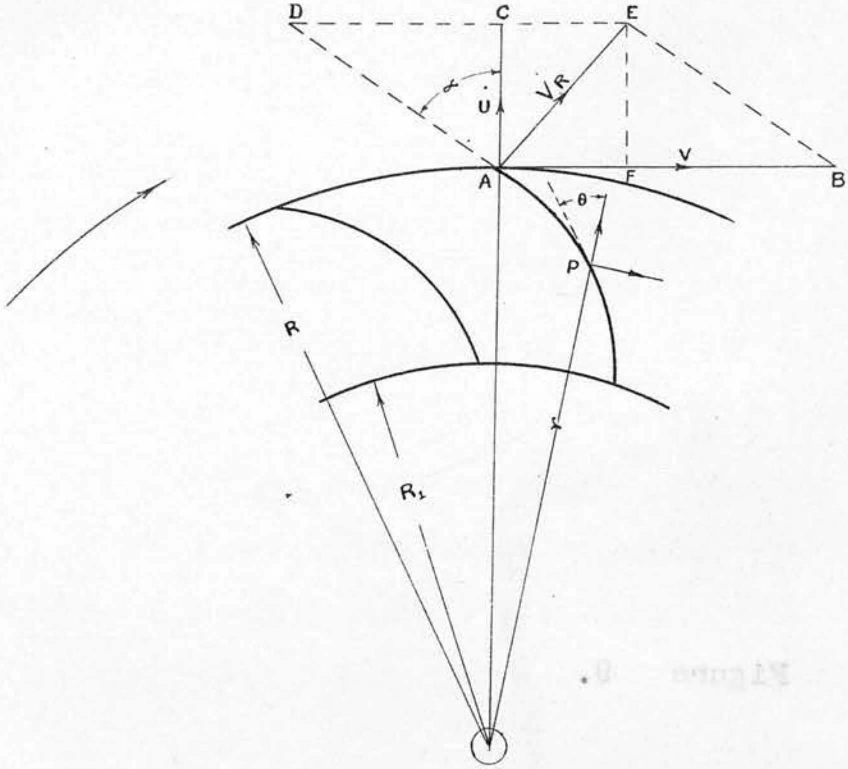

Figure 9. Vectorial Diagram of Velocities fo
Fan with Blades backward-trending
at their outer Extremities.

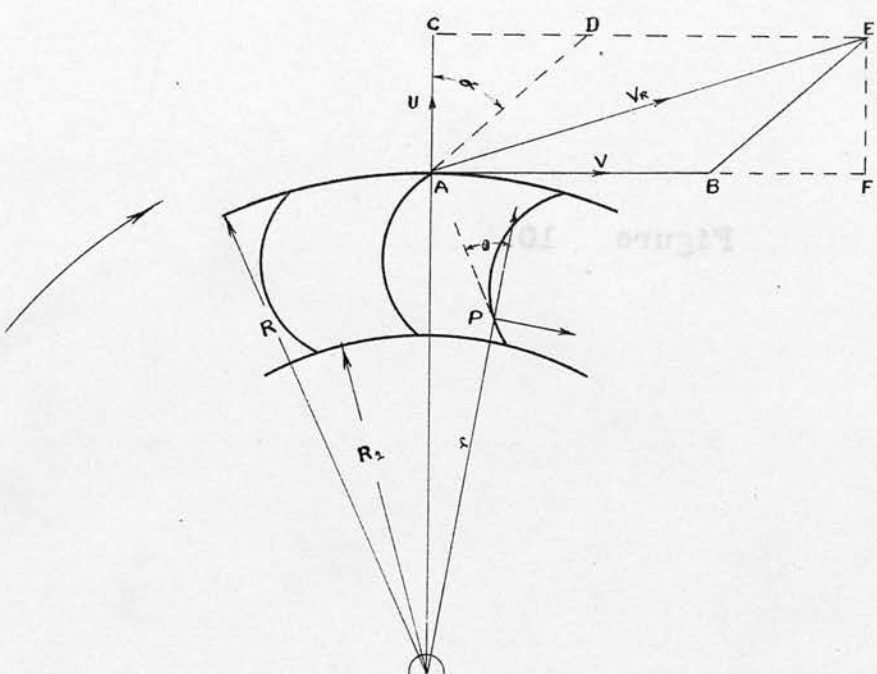

Figure 10. VectorialDiagram of Velocities
for Fan with Blades turned
forward at their outer
Extremities.

is forward, and when given the negative sign, the reverse is inferred. In both diagrams, the magnitude and direction of the radial component of velocity, U, is represented by AC, and the peripheral velocity, V, by AB. The velocity relative to the blade, Vb, is obtained by drawing CD normal to AC until it meets the tangent to the blade at its tip; CD represents this velocity, to scale, and it may be expressed thus:-

$$V_b = U \sec\alpha \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

The resultant velocity of emergence must be compounded from AD and AB, and is accordingly represented in magnitude and direction by AE. From the diagram we observe the reason for the high volumetric yield of fans of the Sirocco type, i.e., with forward-trending blades at their tips; in Figure 10, the resultant velocity, V_R, is greater than V, the tangential velocity, whereas, with fans having their blade tips turned backward as in Figure 9, the opposite is usually the case. Assuming that the air passes through the fan without turbulence, it is also evident that the blades are responsible for the creation of the tangential component, CE, and since,

$$CE = AB \pm BF \qquad \text{we have:-}$$

Tangential Velocity of the Air at the instant of emergence $= V \pm U \tan\alpha \ldots (4)$

(b) Work done on the Air in its Passage through a Fan.-

One of the most important of the mechanical laws applicable to fans is that the change of moment of the momentum of a mass acted upon by forces is equal to the moment of the impulse of the external forces, or to their angular impulse.[1] The following application of /

1. "The Fan", by C. H. Innes (1916); page 16. (Technical Publishing Co. Ltd).

of this law is due to Professor A. Rateau.[1]

Let q cubic feet of fluid be delivered by the machine per second and the weight of one cubic foot be w lbs. Then the mass delivered per second $= \frac{qw}{g}$.
Take any point, P, (Figures 9 and 10) located on a blade, at a distance r feet from the centre of the fan; let its radial and circumferential velocities be represented by u and v respectively, and the angle of the blade at the point be θ. From equation (4) we have:-

Tangential velocity of air at P $= v + u \tan\theta$.

Tangential momentum of air at P $= \frac{qw}{g}(v + u \tan\theta)$

Moment of momentum about the fan centre $= \frac{qwr}{g}(v + u \tan\theta)$

Total Moment of Momentum about the fan centre $= \sum \frac{qwr}{g}(v + u \tan\theta)$

= Moment of Resultant Momentum

$= \frac{qwR}{g}(V + U \tan\alpha)$

Work done on Air = Total Moment of Momentum × Angular Velocity (ω).

$$= \frac{qwR}{g}(V + U \operatorname{Tan}\alpha)\omega.$$

And, since $\omega = 2\pi n$ and $V = 2\pi Rn$, where n = number of revolutions of fan per second, we have:-

Work done on Air by Fan $= \frac{qw}{g}(V^2 + UV \operatorname{Tan}\alpha)$ ft. lbs/se ....(5

(c) The Theoretical Depression.- (H)

In the perfect fan under discussion, the whole of the energy evaluated by equation (5) would be utilised in doing useful work - none would be wasted, it would all be expended in giving pressure energy to the air./

1. "Sur la théorie des turbines, pompes et ventilateurs", par A. Rateau; presentee par H. Léauté. Comptes Rendus de l'Academie des Sciences, 1896. CXXII, 1268.

air. Hence the fan would support an air column, H feet in height, equivalent to a pressure of Hw pounds per square foot, and when it is delivering q cubic feet of air per second we have:-

Work done on Air by Fan $=$ Hqw ft. lbs. /sec.....(6)

Therefore, from equations (5) and (6),

$$Hqw = \frac{qw}{g}\left(V^2 + UV \operatorname{Tan}\alpha\right)$$

$$\text{or } H = \frac{\left(V^2 + UV \operatorname{Tan}\alpha\right)}{g} \quad \ldots\ldots\ldots\ldots\ldots\ldots(7)$$

This equation gives the correct formula for use in the determination of the depression which a theoretically perfect fan can create. From this, the so-called "theoretical water-gauge"[1] can be calculated by simply converting the units of H (i.e., feet of air column) into inches of water column, thus:-

$$\text{The theoretical water-gauge} = \left(\frac{12\ Hw}{w_1}\right) \text{ inches}$$

where $W_1 =$ weight of a cubic foot of water.

Or, taking $w_1 = 62.4$ lbs. the theoretical water-gauge,

$$(G) = \frac{Hw}{5.2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(8)$$

With fans having their blades inclined forward, the second term in equation (7) is positive; it becomes negative in the case of fans with backward-trending blades, and vanishes when the blades are radial at their tips, the formula in the last case becoming

$$H = \frac{V^2}{g},$$

as given in equation (2).

With the latter type of fan, i.e., radially bladed, the value of H is independent of the volume of air discharged, /

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1. "Water-gauge" is the existing method of measuring the depression; instead of expressing this in feet of air-column, the water-gauge method expresses it in inches of water-column. This latter method will most likely be discontinued in future, and the depression stated directly in "pounds pressure per square foot".

discharged, being affected only by the tangential velocity, V. It is because of this fact that such fans produce their highest depression when sealed off from the mine, *i.e.*, when delivering no air. In the other two classes of fans, the value of H *is* affected by the quantity passing through the fan. In the case of the fan with forward-trending blade tips, the depression is not a maximum when the fan is delivering no air (*i.e.*, when the fan-drift is shut off); as the drift is gradually opened, and the fan begins to pass air, the value of H will rise because of the second term in equation (7). However, as the quantity passing through the fan increases, the resistance of the fan increases, so that, ultimately, the additive effect of the forward-blade-tip curvature is neutralised and the depression begins to fall. Obviously, in fans with backward-trending blades at their outer extremities, the depression will decrease as the quantity of air which the fan is delivering increases. These features are illustrated by the characteristic curves given in Figure 11.

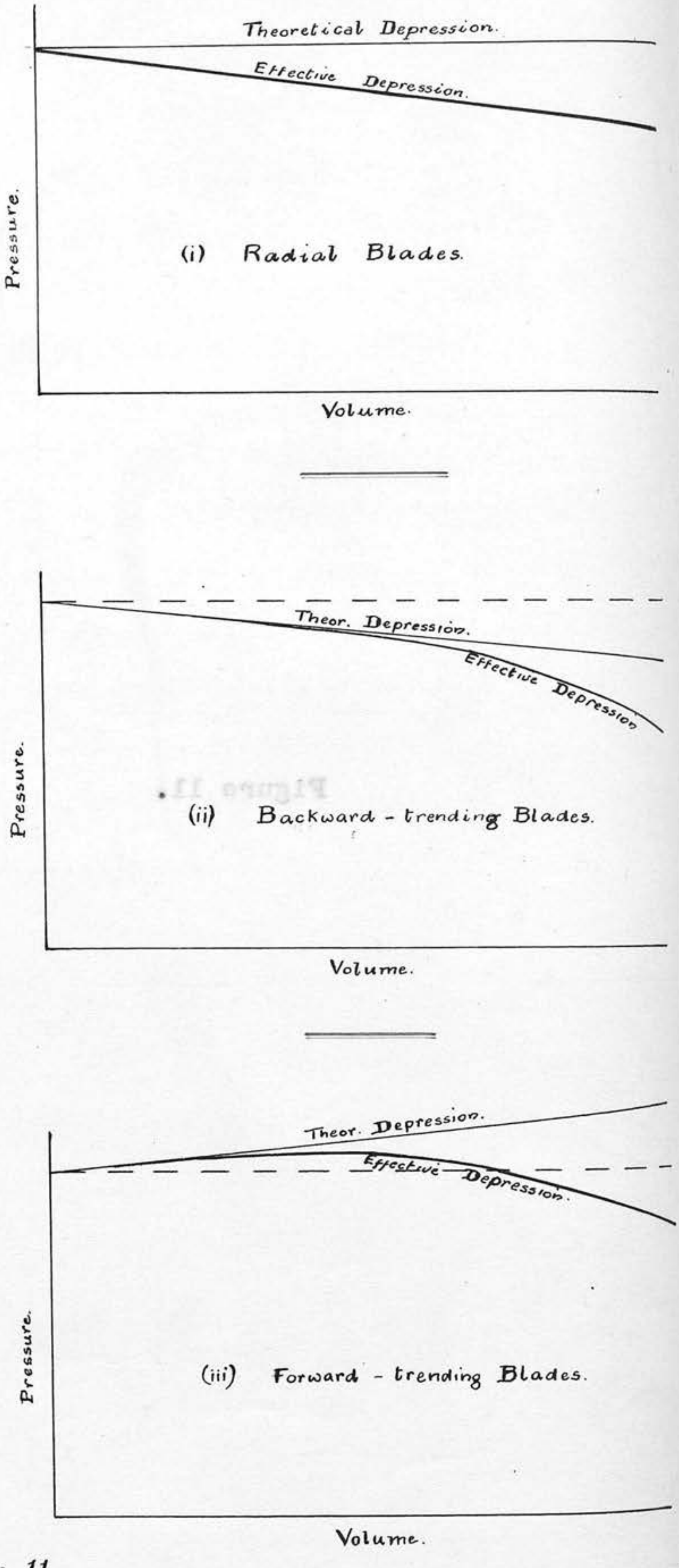

Figure 11. Pressure - Volume Characteristic Curves of Fa
running against a Constant Resistance.

The Actual Ventilator.-

In the foregoing discussion of the hypothetically perfect ventilator, we considered the case where the theoretical depression, H feet of air column, was created when delivering a volume of q cubic feet per second. Coming now to the actual fan, when delivering the same volume, it produces a depression less than H, because of imperfections in its design. Let H_A be the actual depression; this will be measured in the fan-drift by the water-gauge, the connection between H_A and the effective water-gauge, h, being:-

$$H_A = \frac{5.2h}{w} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The loss of depression or head, $H - H_A$, is thus an overall assessment, in terms of "head", of the imperfections in the design of the fan and its adjutages- an assessment which we will make use of in the section dealing with fan efficiencies. This loss of head is due to two principal causes:-

1. Shock, in consequence of the sudden changes in direction, or velocity, to which the air is subjected in its passage from the fan drift through the fan to the external atmosphere.

2. Friction, both between the air and the fan material, (skin friction) and the air particles themselves (eddying and turbulence).

The compressive action of mine ventilators is normally slight, so that any loss of head due to rise in temperature will also be slight, and, compared with the losses resulting from shock and friction, negligible.

While the overall assessment in loss of head is convenient for many purposes it does not lend much assistance in seeking to know where improvements in the design are most required in order to reduce the loss. To be of any practical use, the various sources of loss must be investigated one by one. In the analysis of /

of loss of head which we are to attempt, parallel sided fans only will be considered. Also, for further simplicity, the general assumption that pressure is proportional to the square of the velocity will be made. (The correct index of V, or what amounts to the same thing, Q, the quantity, is one of the important points which the Committee on Ventilation hope to determine).

Losses in Detail.-

1. Skin-frictional Loss between Water-Gauge Position and Fan Inlet. (l_I)

The extremity of the water-gauge is usually placed some little distance from the fan inlet and consequently there will be a slight loss, (l_1) due to skin friction. This loss will be proportional to the square of the air velocity in the fan drift, but since, with any given fan, this velocity is proportional to the effluent radial velocity, U, from the fan, we may write,

$$l_I = k_I U^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (10)$$

where k_I is a constant.

2. Loss due to Sudden Change in Direction of Air at Fan Inlet. (l_2).

It frequently happens that before the air can enter the fan inlet it has to take a sharp right-angled bend. This feature, from a practical point of view, is almost unavoidable with double-inlet fans, but there should be no need for it with ventilators of the single-inlet type; in the latter case, the fan drift should lead directly into the fan inlet. From theoretical considerations, it can be shown that the loss of energy due to sudden bends is proportional to the square of the velocity, and recent experimental work in /

in this connection certainly confirms this.[1] As before, the velocity of air entering the fan is proportional to U, the radial velocity of outflow, so that, for any given ventilator, this second loss of head, l_2, may be expressed thus:-

$$l_2 = k_2 U^2 \quad \text{.........................(11)}$$

where k_2 is a constant.

3. Loss due to Sudden Change in Direction of Air on Entering Fan. (l_3).

In most designs, the air has to take a right-angled bend immediately it enters the fan so as to assume radial flow towards the blades. Where such is the case it involves a loss (l_3) due to shock, similar to (l_2). Or,

$$l_3 = k_3 U^2 \quad \text{......................... (12)}$$

where k_3 is a constant.

With the Rateau fan, an exceptionally well-designed French fan, in which the air is guided gradually into the radial path, the constant, k_3, will be small indeed, and mostly due to skin friction. The loss will also be somewhat modified in the Schiele and Ser ventilators, the hading of the casing sides towards each other in the former, and the larger diaphragm of the latter, tending towards reduction in the value of k_3.

4. Loss due to Shock on Air Entering Fan Blades (l_4).

Figure 12 shows a portion of a fan runner; β is the angle which the inner edge of the blades makes with the radius, V_1 is the tangential velocity at this point, and /

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1. See "Experiments on the Flow of Air in Mines" by Messrs D. & J. Penman. Trans. Inst. Min. Engs., (1924). Vol. LXVIII, page 157.

and U_1 the velocity of the air - assumed purely radial - at the moment of entry. The vectorial diagram of velocities relative to the portion under consideration is given in Figure 13 which is similar to Figure 9. The head lost, l_4, due to shock because the velocity of the air has been suddenly changed from U_1 to Vr is respresented by $\frac{BC^2}{2g}$. But $BC = CE - BE = V_1 - U_1 \operatorname{Tan}\beta$ Hence,

$$l_4 = \frac{1}{2g}\left(V_1^2 - 2\,U_1V_1 \operatorname{Tan}\beta + U_1^2 \operatorname{Tan}^2\beta\right) \ldots\ldots (13)$$

In any given fan, however, the angle β will be constant, and V_1 and U_1 proportional to V and U respectively, hence

$$l_4 = k_4V^2 - k_5\,UV \quad k_6U^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

where k_4, k_5 and k_6 are constants.

As a corollary, it follows from equation (13), that l_4 will be zero when $V_1 = U_1 \operatorname{Tan}\beta$. In other words, for the air to enter the blades without shock,

$$\operatorname{Tan}\beta = \frac{V_1}{U_1} = \frac{2\pi R_1 n}{\dfrac{q}{2\pi R_1 b}} = \frac{4\pi^2 R_1^2\, bn}{q} \ldots (15)$$

It is to be noted, however, that with fixed blades, the angle β can only be determined for one particular volume of discharge. When departure is made from this volume, then the air must suffer shock on entering the blades. Indeed, it is extremely doubtful whether this loss, l_4, can ever be entirely eliminated since, as we shall show experimentally, the air flow into or out of the wheel is not uniform, and therefore U is variable. In most British fans, little attention seems to have been given to this important point, and consequently l_4 is needlessly high. Again, the Rateau fan provides an example of very careful design in this respect.

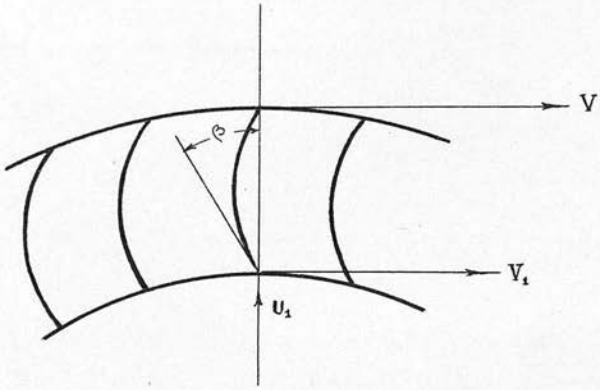

Figure 12. Section of a Fan Runner.

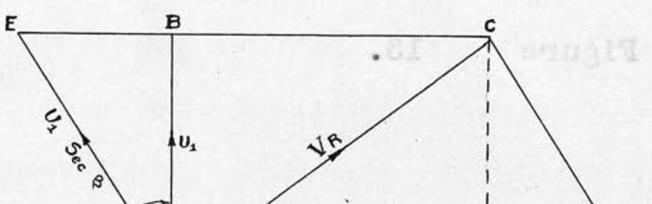

Figure 13. Vectorial Diagram of Velocity.

5. Loss due to Friction in Passage of Air between the Blades. (l_5)

In this case, head will be lost in consequence of both skin and eddy friction. To assess the relative value of the loss of head, (l_5) due to eddying between the blades, presents difficulties. It will simplify matters, and at the same time should not be far from the truth, if we assume such loss as we are presently investigating to be proportional to the square of the velocity relative to the blades (V_b), and to the length of the air passage between the blades.

From equations (2) and (3), $V_b = \frac{q}{2\pi R b} \sec\alpha$ where V_b is the velocity of air relative to the blade at the instant of emergence. Let u be the radial component of velocity at any radius, r, where the angle between the tangent to the blade and the radius is θ; also, let v_b be the component of velocity relative to the blade at the point considered. Then

$$v_b = u \sec\theta = \frac{q}{2\pi r b} \sec\theta$$

The relation between U and u is $u = \frac{UR}{r}$, and hence we can express V_b in terms of U, thus:-

$$v_b = UR\frac{\sec\theta}{r}$$

Consider now a small length of the blade, ds; the loss of head along ds will $= U^2R^2 \frac{\sec^2\theta}{r^2} ds$, and the total loss of head will then be:-

$$l_5 = U^2R^2 \int_{r=R_1}^{r=R} \frac{\sec^2\theta}{r^2} ds$$

The above integral, whatever it may resolve into, is of a character dependent on the dimensions and design of the fan, and in any given case, will be a constant, so that the above expression may be re-written,

$$l_5 = k_7 U^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (16).$$

From the above, it cannot be inferred that l_5 will be small in fans with few blades. While, in such cases, the portion of the loss due to skin friction will be reduced, eddying losses (due to air-slip or re-entry) will be greater.

6. Loss due to Destruction of Radial Momentum. (l_6).

At the instant of emergence from the fan wheel, the radial component of the air velocity is U. In nearly all ventilators, the air is delivered directly into the casing, which results in the destruction of its radial momentum, the value of which loss, l_6, $= \frac{U^2}{2g}$. Since, however, the radial outflow is never uniform, it is preferable to write,

$$l_6 = k_8 U^2 \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (17)$$

In British interests, it is regrettable that once again we have to cite the Rateau fan as an example of sound design in an attempt to minimise this source of energy loss, (l_6). This fan is provided with a "diffuser", (see Figure 14), which is either parallel-sided, or diverges very slightly, and usually forms a spiral about the fan runner. When the diffuser is of uniform width, the moment of momentum of a particle of air through it is constant - that is, the velocity at any point varies inversely as the radius[1]. The only British fan in the design of which an attempt is made to combat this loss (l_6) is the Waddle; in this ventilator, which has no fixed casing as have the other types of centrifugal machines, the blade tips discharge the air into the évasée arrangement shown in Figure 7.

The diffuser is an important addition to the centrifugal ventilator, since its function is to reduce a loss which the évasée comes too late to reduce.

7. Loss due to Partial Destruction of the Tangential Momentum.

Depending on the design of the casing and on the uniformity /

1. Vide Innes, op. cit., page 55.

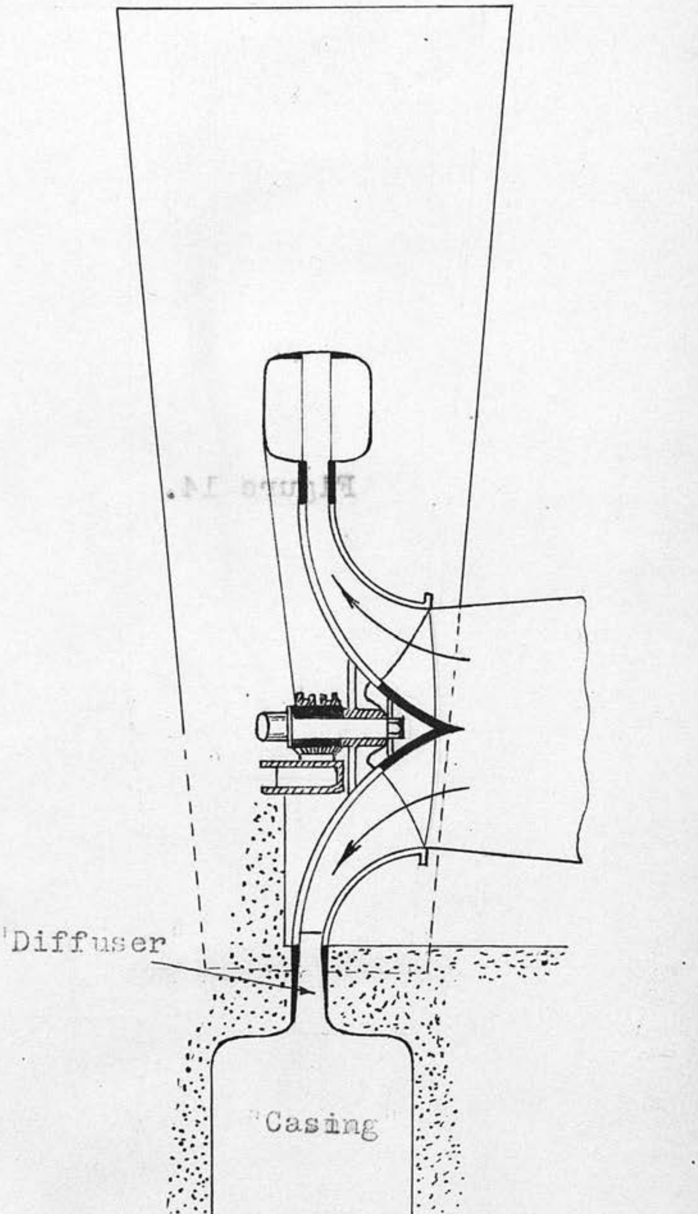

Figure 14. Section of a Rateau Fan.

uniformity of air flow entering it, there will be a loss of head due to the partial destruction of the tangential momentum in consequence of shock and eddying. From equation (4), the tangential velocity of the air at the instant of emergence $= V + U \tan\alpha$, and the resulting loss, l_7, will be proportional to the square of this, i.e.,

$$l_7 = k_9 \left(V^2 + 2\,UV\tan\alpha + U^2\tan^2\alpha\right)$$

Or, since for any given fan, $\tan\alpha$ will be constant,

$$l_7 = k_9 V^2 + k_{10} UV + k_{11} U^2 \quad \ldots\ldots\ldots\ldots(18).$$

The sign of the constant, k_{10}, will be governed by the shape of the blades at their extremities. Hence, other things being equal, l_7 will be less in the case of fans with backward trending blades, than in fans with blades curved forward in the rotational direction.

8. Loss due to Skin Friction in Casing and Evasée. (l_8).

This loss will be proportional to the volume discharged, which, in turn is proportional to U, so that,

$$l_8 = k_{12} U^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(19).$$

9. The Final Loss - due to Velocity of Discharge. (l_9).

In our perfect ventilator, air is discharged with zero velocity; in practice, obviously, this ideal is unattainable. If V_d is the mean velocity of discharge, the head lost is generally taken as being $= \dfrac{V_d^2}{2g}$ the kinetic energy possessed by the air at this stage. As will be shown in the experimental section dealing with évasée design, the air is never discharged with uniform velocity; actually, the velocity is most variable. Since the kinetic energy destroyed at the discharge is proportional to the square of velocity, a greater loss is sustained where uneven flow exists than where the velocity /

velocity is uniform[1]. Hence, the loss we are seeking to assess will be greater than given by $\frac{V_d^2}{2g}$, or

$$l_9 = C\frac{V_d^2}{2g},$$ where C is a constant greater than unity.

But, as before, Vd will be proportional to U^2, so that,

$$l_9 = k_{13}\,U^2 \qquad \text{..................(20).}$$

As the correct design of the évasée forms a considerable part of the experimental work carried out, remedial comments in the reduction of the loss, are consequently reserved for later discussion.

Summation of the Losses (L).

The total loss of head (L) between the points considered will be the sum of the losses represented by l_1, l_2, l_3, etc. Or,

$$L = U^2(k_1 + k_2 + k_3 + k_6 + k_7 + k_8 + k_{11} + k_{12}) + UV(k_{10} - k_5) + (k_4 + k_9)V^2$$

Or, simply,

$$L = K_1V^2 + K_2UV + K_3U^2 \qquad \text{..........(21)}$$
$$= H - H_A$$

As already pointed out, the sign of k_{10} is dependent on the shape of the blade tips. With backward curved tips, k_{10} is negative and hence K_2 is of like sign. When the blade tip curvature is in the rotational direction, k_{10} is positive and K_2 will generally be positive.

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1. For proof of this statement, see "An Experimental Study of Fan Evasées", by Prof. Henry Briggs and the writer; Trans. Inst. Min. Eng. 1925, Vol. LXVIII. Appendix B, page 343.

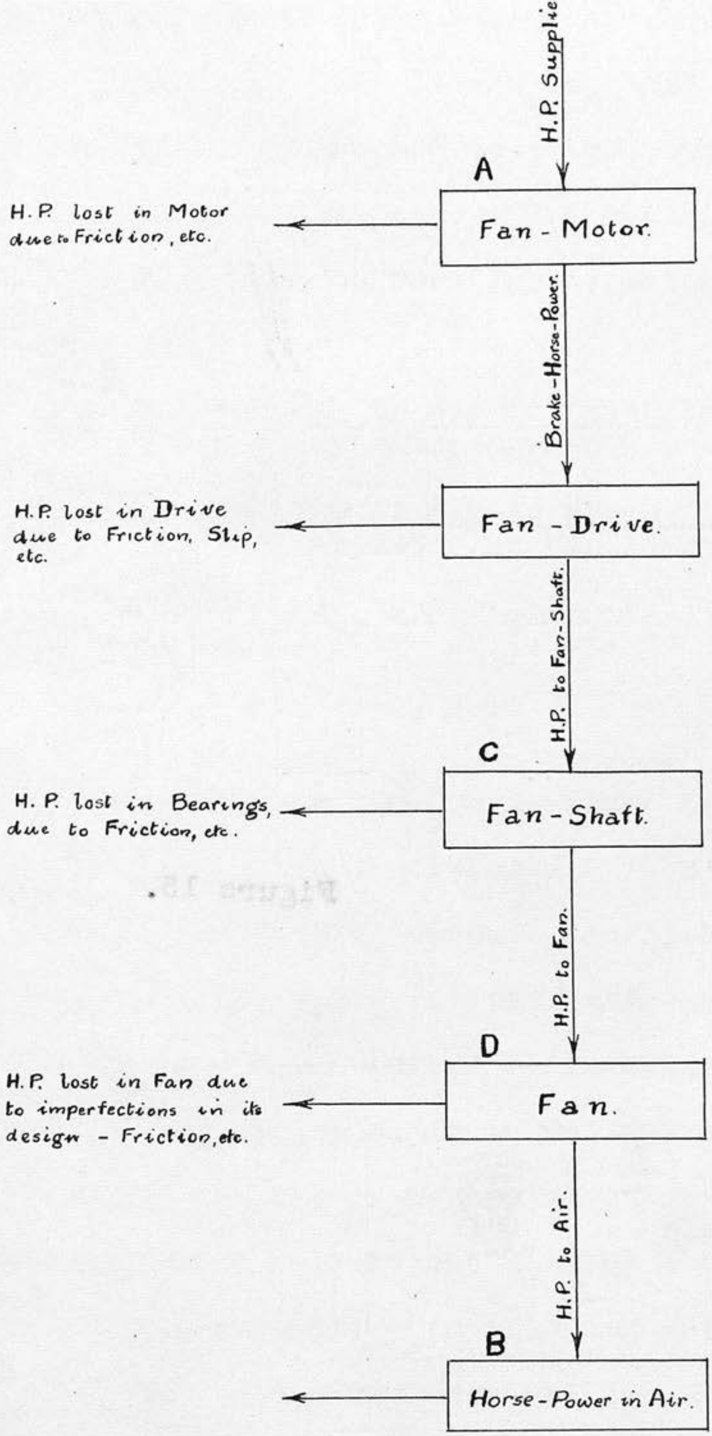

Figure 15.

"Flow-Diagram of Power."

FAN EFFICIENCIES AND RELATIONSHIPS.

As a natural sequence to the many weaknesses in the existing tenets, the subject of fan efficiencies is also somewhat clouded. For example, we find differ:
:ent writers giving different definitions of certain efficiencies, all of which add to the bewilderment of the mining student. In Figure 15, what may be termed the "Flow Diagram of Power" is given, together with the various losses involved. The efficiencies, (i.e., ratio of work output to work input) which concern us most are:-

(a) $\frac{B}{D} = \frac{\text{Horse-Power in Air in Fan-Drift}}{\text{Horse-Power entering Fan}}$ = Manometrical Efficiency.

(b) $\frac{B}{C} = \frac{\text{Horse-Power in Air in Fan-Drift}}{\text{Horse-Power entering Fan Shaft}}$ = Mechanical Efficiency.

and (c) $\frac{B}{A} = \frac{\text{Horse-Power in Air in Fan-Drift}}{\text{E.H.P. of Motor or I.H.P. of Engine}}$ = Overall Efficiency or "Useful Effect".

To evaluate the "Horse-Power in Air in Fan-Drift", i.e. the total energy possessed by the air at that part of the ventilating circuit, involves two fundament:
:al measurements, viz:- (1) effective depression, and (2) volume passing. Before discussing the above efficiencies in extenso, it is perhaps advisable to consider the methods of assessing these two factors, and hence arrive at some decision as to which are the most accurate methods. In this connection, much controversial matter has been written, and much other:
:wise careful experimental work rendered practically valueless simply for the lack of recognised standard methods. A perusal of the literature on ventilators and ventilation which has appeared from time to time in the Transactions of the Institute of Mining Engin:
:eers will occasion surprise regarding those fundamental measurements; /

measurements; truly, they have been veritable scapegoats in past discussion. Fortunately, there is now reasonable hope that the Committee referred to at the outset will issue a dictum on this all important question.

1. Measurement of the Effective Depression.-

The recognised method of obtaining the effective depression, as distinct from the theoretical depression (already sufficiently discussed) is the well-known water-gauge, which measures the required "head" in inches of water-column. That the "inches of water column" method of expressing the "head" is about to be abandoned and replaced by the much more sensible one of "pounds per square foot", has been previously mentioned; the change will only involve the regraduation of existing manometers.

There are three distinct forms of "pressure head" which can be measured in the air passing along the fan-drift, viz:-

(i) Velocity "Head". This is the "pressure head" necessary to accelerate a mass of air from a state of rest to the final velocity it attains - the kinetic energy possessed by the air.

(ii) Static "Head". This "head" overcomes the resistance to airflow and is the pressure which would be measured by the difference between two barometers, one placed inside the fan drift, and the other in the external atmosphere.

(iii) Dynamic or Total "Head". This is the pressure required to overcome the resistance to flow, and to create the velocity of flow. It is the algebraic sum of (i) and (ii).

Our immediate object is to measure this last "head" - the "head" due to the momentum of the air-current.

Figure 16 shows the extremities of several forms of pressure gauges in direct connection with the fan /

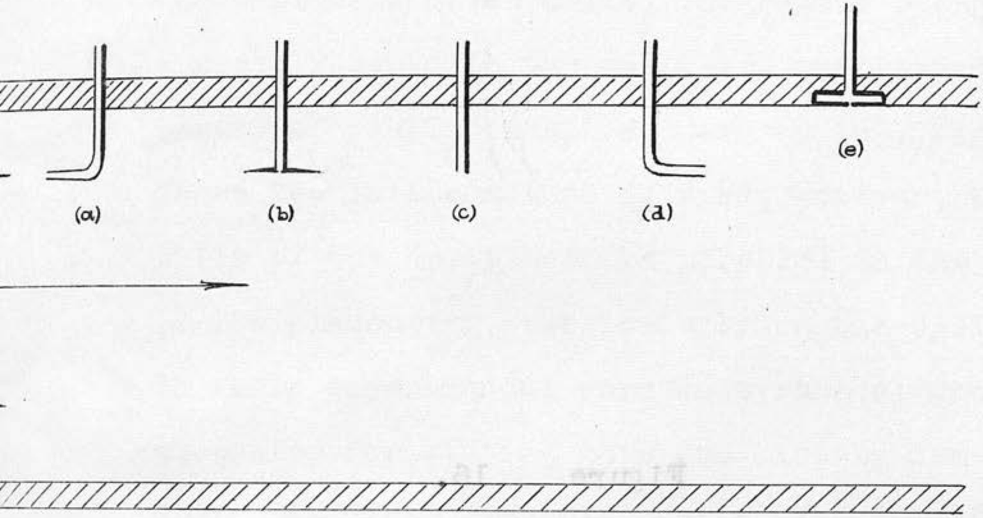

Figure 16. Forms of Pressure-gauge Extremities inFan-drift.

fan drift. It is now widely recognised that the form (a), i.e. facing upstream, is the correct one to use in assessing the value required.[1] Nevertheless, there are still some authorities who contend that a modification of the form (b) is also correct.[2] When fitted with the Darcy or Dupré tip, (b) will register the static pressure; or such pressure may be even more accurately determined by the use of the piezometer method in which the gauge is connected to a recess in the wall of the fan drift, the recess being closed by a plate flush with the wall, and perforated by a series of small holes, as shown by (e), Figure 16. Where the latter method is used, however, the opposite walls of the drift must be parallel in the vicinity of the piezometer, particularly on the upstream side. In early experimental work on pyramidal ducts, the necessity for stating this was clearly demonstrated. Where the sides diverged, the static "head" registered by the piezometer was too large because of the suction effect at the small holes (they were only $3/_{64}$th inch in diameter), while the opposite was the case when the piezometer was tried in a converging duct. In the latter instance there would be a slight conversion of velocity energy into pressure energy taking place because the plane of the mouths of the piezometer holes was inclined towards the axis of the air stream, the conversion being the greater, and hence the "head" recorded /

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1. Eg. See "Mine Ventilation" by Prof. D. Hay. Coll. Managers' Pocket Book (1924) page 102.
2. Eg. See "The Measurement of Pressure, with special Reference to the Testing of Fans", by T. Bryson. Trans. Inst. Min.Eng. (1915) Vol. XLVIII, page 50.

recorded the smaller, the greater the convergence of the duct.

The shape or size of the dynamic or impact tube (a) does not matter[1] as long as its axis is parallel to the axis of flow. Regarding the latter condition, however, any small errors in setting the direction of the tube have no appreciable influence.[2]

The form shown by (d), but "sleeved" in flannel, was that used by the Joint Committee of the North of England Institute of Mining and Mechanical Engineers and Midland Institute of Mining, Mechanical and Civil Engineers in their fan tests referred to in Part I. As is very evident, this form does not even measure the static "head" correctly, but will record a greater difference of pressure due to an ejector suction effect. The same applies to the form (c) only to a less degree. In passing, it may be mentioned that the latter is the one favored by fan makers when testing the performance of exhaustive ventilators, for very obvious reasons. For instance, in fan-drifts where the velocity is exceptionally high, very exaggerated depressions will be recorded with this form of gauge extremity. The full significance of this point would be forcibly brought home to those favour:ing the use of gauge tip (c), if tests were carried out on compressive ventilators, and the effective "head" measured where the velocity was high.

A most important matter in connection with the measurement of the effective depression is the position of /

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1. Vide E. A. Griffiths in "Engineering Instruments and Meters" (1920) p. 96.
2. Ibid, page 97.

of the dynamic tube in the fan drift. While the static "pressure" is uniform over any cross-sectional area, the dynamic "pressure" is, due to the differences in velocity at various points of the cross-section, very variable. In preliminary experimental work with the dynamic tube this variation was so emphatically demonstrated that it became necessary to determine the position inside a duct at which the impact tube would register the mean value for any cross-section considered. The required position was found to be at a point one-seventh of the width of the duct or gallery measured from either side, along a central axis. (See Appendix A).

Regarding the best type of manometer for use, there are varied opinions. Certainly, the ordinary vertical U-tube in general use at mines, while perhaps good enough for local comparative purposes, such as checking the daily working of the fan, is hopeless for absolute determinations. It is indeed doubtful whether its accuracy can approach a reading of 0.05 inch of water column (i.e. 0.25 pound pressure per square foot). Oscillations of the surface of the liquid employed (usually water) cause difficulty in reading, and many attempts to eliminate such oscillation by introducing resistance at the bottom of the tube resulted in a decrease in the sensitivity of the instrument. Several micromanometers on the differential principle have been introduced from time to time, but these are chiefly intended for Laboratory purposes. One of the latest of this class of instrument is the Wahlen Micromanometer[1] for which is claimed /

1. "Wahlen Micromanometer", by Louis W. Huber; "Coal Age", Vol. 23, No. 17, (1923).

claimed a sensitivity as high as 0.0001 inch of water column (i.e., a "head" = 0.0052 pound pressure per square foot). However, when the possible degree of accuracy attainable in the measurement of other factors (e.g., volume passing) is considered, it is perhaps taking matters to too fine a limit to use such an instrument as just mentioned.

The manometer used in all my experimental work in the Mining Laboratory of this University consists of an inclined glass U-tube, 5 feet long, 3/4-inch in diameter, and graduated in tenths of an inch. Instead of water, petrol was used (sp. gr., 0.758), which, besides keeping the inside of the tube clean and giving a well defined meniscus, added to the sensitivity of the instrument. It was inclined at 7°35' which gave a multiplication factor of 0.1 when converting the inclined measurements into the desired vertical measurements. The inertia of the comparatively large volume of liquid materially assisted in minimising the oscillatory effects due to fluctuating flow. With this form of manometer, it is quite easy to read, accurately, pressure "heads" as low as 0.002 inch of water column, (i.e., 0.0104 pound pressure per square foot), which is probably as precisive as need be.

In discussing the First and Second Reports of the Midland Institute Committee on the Ventilation of Mines,[1] Professor Henry Briggs, University of Edinburgh, put forward a strong plea for an early report upon "the all important question of taking water /

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1. Trans. Inst. Min. Engs., 1924; Vol. LXVII, p. 303.

water-gauge readings". He rightly contended that the settlement of so fundamental a point should be a primary consideration. It is satisfactory to be able to state that such a Report is now <u>sub judice</u> and is likely to emanate from this University.

they must be handled accordingly. The necessity for their reliable calibration at least every six months cannot be too strongly urged as their accuracy decreases with use and more so with misuse. The range of usefulness of such instruments, where accuracy is desired, is limited, as indicated above. For low velocities, such as usually prevail at and near the working places, anemometers are hopeless, and it is here that the kata-thermometer, recently devised by Dr. Leonard Hill, may find a sphere of practical utility. Turbulent and pulsating flow renders the accuracy of wind-vane air meters impossible owing to the inertia of their rotating part; they cannot be calibrated for quickly changing puffs of air.

Nevertheless, provided the conditions enumerated above are given the attention they require, the mean value of several readings cannot be far wrong. A 6-inch Biram anemometer, which has the advantages of being self plumbing, and a 2½ inch Davis zero-setting anemometer, were greatly and satisfactorily used in the experimental work subsequently to be described, both instruments being frequently calibrated by means of the "anemometer table"[1] in the Laboratory of the Mining Department.

The Pitot tube, as a reliable velocity measuring instrument, is, like the anemometer, not without its critics. Many forms of this tube have been used, but the one mostly favoured, since it has unity factor, is that produced by the Cambridge Scientific Instrument Company[2]. One of similar design was /

1. See "The Testing of Anemometers" by James Cooper. Trans. Inst. Min.Eng., (1921-1922). Vol. LXII, pp.91 et seq.
2. See "Experiments on the Flow of Air in Mines", op.cit page 160.

was made and was used in most of the experimental work, with excellent results. One serious drawback, however, discovered with this and similar instruments, where the static holes are of exceedingly small diameter (in our case, the diameter of these holes was 1/50th inch) is that it is soon rendered inoperative where there is moisture in the air, as a film of water readily forms over the static orifices. A similar source of trouble in the use of the Pitot tube is recorded by Mr. F. A. Steart in his recent paper, "The Application of Air-Screws to Mine Ventilation".[1]

The accuracy of the Pitot tube is also very poor with low air velocities, due to the extremely small pressure heads which such velocities record - one inch of water-column is equivalent to a velocity of approximately 4000 feet per minute. Barometric and hygrometric observations in the fan-drift are necessary with this method of measurement, in order that the weight of a cubic foot of air may be determined. The ordinary wet and dry bulb hygrometer is generally used in this connection. Care must be taken, however, not to expose this instrument to high velocity air; if this is unavoidable, then the bulbs should be covered with wire gauze which will protect them from draughts. The Pitot tube method thus involves much laborious calculation where several readings of the gauge are necessary, although tables covering a practical range of velocity "heads" and atmospheric conditions could be readily compiled to eliminate this. Like the simple impact tube, the position of the Pitot tube in the fan-drift, so as to register /

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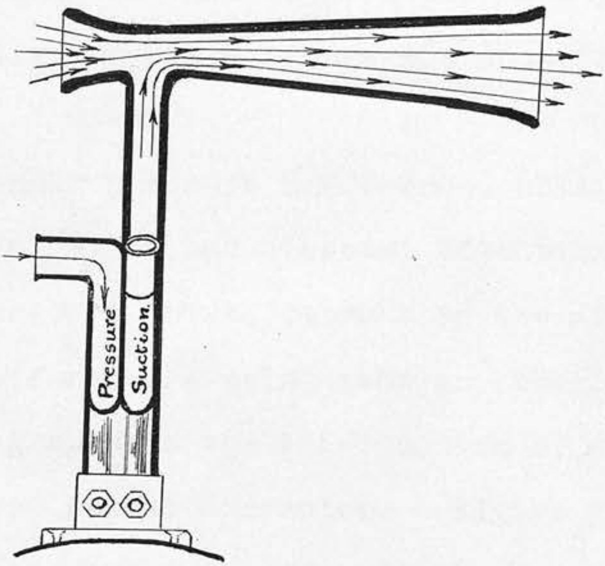
1. Trans. Inst. Min. Eng., 1925, Vol. LXVIII, page 312.



. Figure 17. Venturi-Pitot Pressure-gauge for Measurement of Velocity.

register the mean velocity, must be known. This position has been theoretically proved to be one-eighth of the diameter or width of the duct measured from either side.[1] This position, however, does not agree with that empirically determined by the Prussian Firedamp Commission in 1884; the position of mean velocity was then found to be at a point on the diameter one-sixth across the duct from either side.

The small pressure differences obtained with the Pitot tube method has directed attention to means of increasing these, other than the micromanometer method. Efforts are being made in other branches of engineering towards the introduction of a modified Pitot of a more robust character. Figure 17 shows a type now used by the American Air Force.[2] Due to the Venturi arrangement on the "static" limb giving a suction instead of a static pressure, the readings obtained thereby are considerably amplified. Perhaps a modification of this idea could be advantageously adapted to suit mining purposes.

Hot-wire anemometers, or electrical velocity meters, are a recent development in the measurement of air flow but so far they do not seem to have reached the stage of a reliable direct-reading instrument suitable for mining purposes. There is, however, at least one portable direct-reading hot-wire device on the market, claimed to be specially suited for the measurement of mine air flow; this is the Macgregor-Morris Anemometer[3] made by the Cambridge and /

1. "Flow and Measurement of Air and Gases", by A. B. Eason, M.A. (Griffith, 1919), page 157.
2. Griffiths (op. cit) page 193.
3 "The Macgregor-Morris Anenometer"; Trans. Brit. Assoc., 1922. SectionA.

and Paul Scientific Instrument Company. The Birmingham University Research Department is at present testing the reliability and practicability of this device, and are shortly to issue a report thereon. One of the disadvantages of this type of instrument is the rapid contraction of the scale with increasing velocity of flow; it is very sensitive indeed at low velocities, but its calibration curve soon flattens out as the rate of flow becomes higher. In the instrument mentioned, it would be difficult to read velocities exceeding 400 feet per minute, which is too low an upper limit for mine purposes. For exploration work inside a fan casing, where the air velocities are extremely high, Professor Briggs designed a form of hot-wire anemometer specially suited for the measurement of both rapid rates of air flow and direction of flow. This device will be discussed in the record of experimental work.

The modern desire is for an accurate form of recording flow-meter which will replace the methods of measurement just discussed. In other spheres of engineering, recording instruments are being introduced wherever possible so that those concerned are in a position to check results frequently with the minimum of trouble. Undoubtedly, the use of such recording appliances has focussed attention on inefficiencies in numerous instances and have thereby more than justified their installation. The intro:duction, then, of a practical type of recording flow-meter, with a high degree of accuracy, cannot come too soon. Regarding the remaining measurements required before the three efficiencies referred to can be determined, there is little or no conflict of opinion./

opinion. To estimate the manometrical efficiency it is necessary to know the weight of unit volume of the air which is passing through the fan. The manner of ascertaining this value, and the precautions necessary, have been already discussed in connection with the Pitot tube. Another measurement required before the manometrical efficiency can be assessed is the speed or velocity of the fan. The tachometer is the best instrument for such measurement, especially for modern quick-running fans. The determination of the horse-power entering the fan shaft — required in the assessment of the mechanical efficiency of the fan — presents a difficult practical problem. In the laboratory it can be done by cutting the shaft in two and measuring the torsional effort of the shaft. In practice, however, we have to be satisfied with a brake-horse-power measurement taken at a point which would eliminate, as far as was practicable, all likely sources of loss between the prime-mover and the fan shaft. If the fan is electrically driven, then the total power supplied to the plant is easily obtainable; if a steam engine is used, the total power input must be determined by means of some reliable form of indicator.

Manometrical Efficiency.

The manometrical efficiency of a ventilator is supposed to be an abstruse ratio, difficult to determine, and its practical utility even denied by some writers. It would therefore seem necessary to enquire into this matter, ab initio, moreso, as in the writer's opinion, the manometrical efficiency is the true efficiency of a fan and provides the best criterion as to the suitability of design. The con: :fusion prevalent is perhaps less surprising when we realise that there exist two definitions for this efficiency. (Also, see under "Aerodynamical Efficiency"). The first of these definitions we may express thus:-

$$\text{Manometrical Efficiency (M)} = \frac{\text{Initial Depression}}{\text{Theoretical} \quad ''}$$

the "initial depression" being determined by completely closing the fan drift, and the theoretical depression from the formula, $H = \frac{V^2}{g}$, the air column, H, being converted into the equivalent water-column, since the "initial depression" is so measured. This was the meaning which Murgue attached to the term (op. cit. page 20), and his lead has been followed by more recent writers.

The term "initial depression" requires some explanation. As a sequel to his famous "equivalent orifice" theory, wherein the mine was assimilated to an orifice in a thin plate which would affect the same resistance to the flow of air as did the mine, Murgue introduced the "orifice of passage" theory. In the latter, we have to imagine the fan replaced by another sharp-edged orifice which would effect the same resis: :tance to the air flow as would the fan. To deter: :mine the "orifice of passage", Murgue's instructions were to completely close up the fan drift, (seal it off /

off from the mine) and note the depression created by the fan at a given rate of revolution. This depression, measured in inches of water-column, Murgue dalled the "initial depression" and we may express it by the symbol, h_i. The fan-drift is next opened to the mine, the same fan speed being maintained as when the mine was shut off, and the effective depression, h, observed. Murgue's assumption was that h was always less than h_i, and accordingly the connection between them may be expressed thus:-

$h_i = h + h_o$, where h_o is the difference between the "initial" and the effective depression, sometimes called the "hidden water-gauge" by certain writers. Murgue subsequently deduced the expression,

$$h = \frac{h_i}{1 + \frac{a^2}{o^2}}$$

where a is the "equivalent orifice" of the mine and o the "orifice of passage" of the fan, which allows the latter value to be determined.[1] However, the whole of this deduction collapses whenever the "effective depression", h, exceeds the "initial depression", h_i. With fans having their blades turned forward in the direction of rotation, this does occur, as shown in Figure 11, and hence, by using the depression ratio, $\frac{\text{effective}}{\text{initial}}$, we may get the absurdity of an efficiency exceeding 100 per cent. This in itself ought to have been sufficient reason for looking askance at such a method of determining the manometrical effiency, and also the "orifice of passage". In the past, however, manometrical efficiencies over 100 per cent. seem to have been accepted with perfect equanimity, as if the fan /

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1. op. cit., page 5.

fan was a machine endowed with a secret power. Numerous cases are on record which establish the fact that a fan with forward-trending blades can create a higher depression when delivering air against a resistance, than when the fan drift is shut off.[1]

Murgue's "orifice of passage" method as a means of determining the resistance of a fan stands condemned therefore by the touchstone of experience.

We have also seen, in the discussion on the influence of blade curvature, that the formula, $H = \frac{V^2}{g}$, is applicable only to a particular case, i.e., the radially bladed fan. The first definition given for manometrical efficiency is thus irrevelant from this standpoint alone, quite apart from the anomaly of the "initial depression".

Wabner[2] gives a correct definition for the manometrical efficiency, viz:- "On dividing the effective (observed) depression by the theoretical one, the manometrical efficiency is obtained". This defin:iition is the one now generally agreed upon. At the outset, we defined the manometrical efficiency as being the ratio,

$$\frac{\text{Horse-power in air in fan-drift}}{\text{Horse-power entering fan}},$$

and it has yet to be shown that the definition which we have culled from Wabner, conforms to the accepted idea of an efficiency ratio, i.e., a ratio of power given out to power supplied. Wabner's definition we can express symbolically thus:-

$$M = \frac{h}{G} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(22).$$

Multiplying /

1. For example, see "Experiments on Centrifugal Fans", by Bryan Donkin; Min. & Proc. Civ. Eng., Vol. CXXII, - Appendix I, page 277.
2. op. cit., page 179.

Multiplying the numerator and denominator of a ratio by the same thing does not change the ratio, and hence we can re-write equation (22) as under:-

$$M = \frac{\frac{h \times 5.2 \times q}{550}}{\frac{G \times 5.2 \times q}{550}}$$

In this form, the numerator is the actual output of the fan - the horse-power in air in fan-drift - while the denominator is the power which would be usefully expended had the fan no internal resistance. However, in our hypothetically perfect fan of 100 per cent efficiency there is no internal resistance, and hence all the power supplied to it - assessed by the denominator of the last expression - is converted into horse-power in the air. We have therefore a ready means of calculating the power input to the fan, free of all external losses. Further, since there is no practical way of assessing this power, and since it is just the value we most desire to know in the determination of the actual fan's behaviour, the manometrical efficiency is undoubtedly the best criterion of design of the ventilator we possess; it measures the loss of power due to the fan and its adjutages only, and is not mixed up with any losses in the power transmission.

In the most modern development in the theory of the ventilator and ventilation, we can express the manometrical efficiency as a ratio of resistances (in Atkinsons[1]). Professor Briggs, in a recent article[2] wherein he ably demonstrates the practical utility of the /

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1. The proposed definition of an Atkinson is 'that resistance which absorbs 1 pound pressure per square foot when a volume of 1000 cubic feet of air per second is passing'.
2. "Developments in the Theory of Centrifugal Fans", by Professor Henry Briggs, "Coal Age" (1923) Vol. XXIII, page 601.

the new units of resistance advocated by Penman, deals with, inter alia, the manometrical ratio as a means of determining the relation between the fan and the mine. He expresses the manometrical efficiency,

$$M = \frac{R}{R + r} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (23).$$

where R is the resistance of the mine in Atkinsons, and r the resistance of the fan in the same units.

From equation (23) we are able to determine the actual ratio of the total power absorbed by the fan itself. Obviously, the desideratum is to have a fan with a small resistance in comparison with that of the mine, and a high manometrical efficiency affords us the best assur: :ance we can have in this respect.

It can be shown that the manometrical efficiency is constant for a given fan and independent of the fan's speed, provided the mine resistance remains constant[1], and from equation (23) it follows that the fan's resistance is also independent of the speed - an important result.

Mechanical Efficiency of the Fan.-

This efficiency has frequently been confused with the efficiency of the whole plant. To assess the latter efficiency and term it the mechanical efficiency of the fan is somewhat analogous to the old term, "duty of a pump", where the coal consumption is compared with the water lifted. The true mechanical efficiency of a fan is the ratio between the horse-power in air in fan drift and the horse-power delivered to fan shaft. As previously pointed out, the latter power is difficult to measure in practice. However, the only difference between the mechanical and manometrical efficiencies will be due to losses at the fan shaft bearing, which in many cases will be small indeed. All other losses of energy/

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1. op. cit. page 603.

energy between this point and the point of power supply to the motor or engine, are assessed in the

Overall Efficiency (or "Useful Effect" as it is sometimes termed).

While great stress has deservedly been laid upon the value of the manometrical efficiency, since it concerns the efficiency of the fan and its adjutages alone, nevertheless, the overall efficiency, i.e., the efficiency of the whole plant, is of the utmost econom::ical importance as it includes all losses in the ventilating system. The analysis of possible losses in power should actually go beyond the last stage shown in our "flow-diagram of power", although subsequent losses are in some measure reflected in the value of B (Figure 15). Nevertheless, due to excessive leakages in the circulating system, many instances have been cited wherein a huge volume has been passing through the fan, but only a small fraction of such volume has been actually doing useful work,[1] thus involving an enormous expenditure of useless energy.

The orthodox measurement of the horse-power in the air does not require that its assessors go farther afield than the fan-drift, there to determine the ventilating pressure (i.e., the effective depression) and the volume passing. Such a measurement, however, may be delusionary and promote the feeling that all is well in the ventilating circuit, since an ample sufficiency of air-flow may have been measured at what appeared a reasonable depression. (In this connection, also see under "Clive's Work", Part III, re natural ventil::ation)./

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1. Eg. See Mr. Kerr's remarks in the discussion of Mr. J. Parker's paper, "Economy and Efficiency in Ventilation". Trans. Inst. Min.Engs., Vol. LXVII, (1924), page 12. Mr. Kerr records a case where surface leakage amounted to 64 per cent.

ventilation). Investigation at other points in the air circuit would, in many instances, forcibly indicate that much of the ventilative power was being expended to no useful purpose whatsoever. This phase of mine ventilation while perhaps an apparent digression from the subject of our thesis, is nevertheless one which experience shows to be inevitiably bound up in the question of efficient and effective ventilation. As Mr. Parker points out in his excellent paper, "Economy and Efficiency in Ventilation"[1], there is a marked distinction between the terms 'efficient' and 'effective' when applied to mine ventilation. By a thorough search for all likely cases of disadvantageous leakage, and subsequently effecting efficient remedial measures, it may be quite possible to considerably reduce the ventilative power consumption.

An outstanding example of what may sometimes be achieved, in economising in the power required for mine ventilation, and at the same time increasing its effectiveness, has been recorded by Mr. C. E. Stuart[2] who gives an account of the results of ventilating improvements carried out at three separate collieries over periods of from 2 to 3 years. The alterations were preceded by investigations and tests which elicited the following facts:-

(a) the fans were dealing with an amount of air in excess of the mine requirements;

(b) a large proportion of the air failed to reach the working places due to leakage;

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1. <u>Trans</u>. <u>Inst</u>. <u>Min</u>. <u>Engs</u>., (1923-1924) Vol. LXVI, page 14.
2. "Increasing Coal Mine Efficiency", by Chas. E. Stuart, Fuel Administration, Washington, "Coal Age", 1918, Vol. 14, page 774.

(c) the ventilating pressure was unduly high owing to there being too few splits;

and (d) the efficiencies of the engines driving the fans were very low.

These conditions were remedied gradually. Air leakages were reduced, and the speed of air current decreased by (a) creating more splits, (b) cleaning up falls in the air courses, and (c) by the use of additional air courses wherever practicable. Two-speed motors were also substituted for the steam engines previously used. The net result of these improvements may be summarised thus:-

Case I. Power reduced to 1/6th in 3 years.

Case II. Power reduced to 1/4th (approx.) in 3 years.

Case III. Power reduced to 2/5th in 2 years.

During the period of my research, I assessed the overall efficiencies of three modern mine ventilator installations under normal working conditions, during a period in summer when the natural ventilation effect was practically negligible, and briefly summarise the results hereunder:-

Fan	Horse-Power Supplied.	Horse-Power in Air.	Overall Efficiency. (%)
Keith-Blackman	80.5	41.2	51.2
Waddle	173.6	79.4	45.7
Sirocco	198.3	74.4	37.5

While none of these three fans was running at its rated capacity — the volume of air delivered in each case being considered sufficient for the effective ventilation of the respective mines at the time of test — nevertheless, this continuous heavy loss of power in such a hard pressed industry is almost criminal.

Aerodynamical Efficiency.-

The advocates of this efficiency seek to differentiate between losses due to imperfections in design and blade slip on the one hand, and losses due

to /

to the internal friction of the fan on the other.
If H be the theoretical head produced by the perfect machine, and h_m that which it does produce, <u>neglecting internal frictional losses</u>, when running under the same conditions as did the perfect fan, then

$$\text{Manometrical Efficiency} = \frac{h_m}{H}$$

If h_f be the head lost in consequence of purely internal friction of the fan, then,

$$\text{Aerodynamical Efficiency} = \frac{h_m - h_f}{h_m}$$

In thus attempting to split hairs, the question of fan efficiencies is, in our opinion, being made too complicated; and further, it cannot be logically argued that, to do so, will be of any great practical utility. The imperfections in the design are certainly the cause of the chief sources of loss, namely, eddying (in fan and its adjutages) and stream-collisions; the loss due to purely skin friction would be found to be (could it be accurately and separately assessed) a very small portion of the total loss. We therefore consider it preferable to stick to the conception of manometrical efficiency already dealt with under that title, as being the best criterion regarding the efficiency of the actual ventilating machine.

The Committee on Ventilation Theory now functioning will perhaps end the existing confusion regarding the real fan efficiency by authoritatively defining it.

<u>Volumetric Efficiency</u>.-

This can scarcely be termed an efficiency since, with fan having forward-trending blade tips, it exceeds 100 per cent; volumetric ratio is a better term.

$$\text{Volumetric Ratio} = \frac{\text{Volume of air discharged per revolution}}{\text{Cubical capacity of the fan.}}$$

The ratio, however, has little practical value.

Fundamental Relationships.-

Apart from the power relationships just discussed, there is general agreement regarding the fundamental laws connected with the performance of ventilators. The following laws have been firmly established, the proviso being that the mine resistance is kept constant.

1. Where Q is the quantity delivered per minute, and N, the number of fan revolutions in the same time,

$$Q \propto N.$$

2. Since h, the effective head, varies as the square of the quantity (with the reservation already made regarding the truth of the index of Q), from (1) it follows that

$$h \propto N^2$$

3. From numerous experiments, it has been shown that the quantity Q is proportional to the cube root of the horse-power supplied, or $H.P. \propto Q^3$. Again, from (1) it follows that

$$H.P. \propto N^3.$$

In the next part of this thesis, it is proposed to give a resumé of the most important of recent researches in connection with mine ventilators, and further relationships developed will be given therein.

PART III.

A REVIEW OF RECENT RESEARCHES

RELATING TO

THE MINE VENTILATOR.

Introductory.

In the following brief survey of recent work bearing upon the ventilating fan, our chief purpose will be to summarily review the main conclusions arrived at by the foremost workers. While little or no advance in design has been made during the past two decades, much has been accomplished lately regarding the theory and practice of the centrifugal machine. The comprehensive review of these latter aspects, however, while within the range of our thesis title, we do not attempt; nevertheless a critique of such work covers by far the larger part of the present section. Since our own investigations are primarily concerned with design, precedence is given to work of this character, although, as already stated, the field for review, unfortunately, is limited.

Bryan Donkin's Experiments.[1]

Considerable value is attached to the work of Donkin in his experiments on centrifugal fans because of the care and thought he expended on the all important measurements - volume and depression. He tested eleven different types of fan, performing about ten tests on each under similar conditions, the same apparatus and engine being employed throughout. The diameters of the fans ranged from 16 to $25\frac{1}{2}$-inches, and the number and shape of the blades varied greatly. In accordance with the finding of the Prussian Mining Commission (1884), Donkin set the dynamic gauge at a distance two-thirds of the radius from the centre of the pipe. Velocity was measured by means of a Pitot tube, the mean of eight divisional /

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1. Donkin, op. cit., pages 265 - 282.

divisional readings being taken. (The mean of the square roots of the eight readings would have been more accurate).

Donkin's practical conclusions (as a result of his tests) are summarised thus:-

(a) Few British and Continental fan makers carry out practical tests on the efficiency of their products. He was of the opinion that the number, shape and direction of rotation of the blades ~~were~~ frequently guessed at. (In one case, Donkin tested a fan running in the opposite direction to that intended by the maker, and thereby obtained a 25 per cent increase in the vol: :ume discharge the mechanical efficiency remaining the same).

(b) The shape and number of the blades have a considerable effect on the performance of a fan. He considered that between twenty and twenty-five blades gave the best results.

(c) The angle of the blade tips has more eff: :ect upon the pressure of the air than upon the mechan: :ical and volumetric efficiencies. (This is doubtful; see "Influence of Shape of Blades", page 22).

(d) The cleanliness of the vanes is an impor: :tant matter. (He carried out two tests in this connection - one with varnished vanes coated with coal dust, and the other, with the vanes thoroughly clean. His tests showed a volume increase of 10.5 per cent. in the latter test, the mechanical efficiency remaining the same).

(e) Insufficient attention is paid to the fan inlet so as to reduce frictional losses at this point. (He demonstrated the advantage of a well designed inlet, by the experiments, - one with a large bell-mouthed inlet, and the other without it. When the inlet was used a considerable improvement resulted; the quantity was increased by 31 per cent. and the mechanical efficiency by /

by 9 per cent.).

(f) The design of the casing had a marked influence on the efficiency. The frictional loss inside the casing is often excessive.

(g) The fan runner should always be accurately balanced. To reduce journal friction at high speeds Donkin recommended continuous lubrication.

The Experiments of Heenan and Gilbert.[1]

The published record of the experimental work on fans carried out by Heenan and Gilbert in 1895 has been described as "one of the beacons of the ventilation literature of the last generation". Their object was, briefly, to determine the best type of fan. The three forms of blades tested are shown in Figure 18. The order of their efficiency is the reverse of their numbering, i.e., blade No. 3 gave the best results. These workers verified the three laws given under our "Fundamental Relationships" (page 58). The majority of their tests were carried out with Heenan fans (drum type), having volute casings. They also experimented with a conically shaped fan, the widths of which were $7\frac{1}{4}$-inches at the fan centre and $1\frac{1}{8}$-inches at the outlet; the diameter of the fan was 16-inches, while that of the inlet was $5\frac{7}{8}$-inches. Unfortunately, a concentric casing was used with this fan, having a clearance of $1\frac{1}{8}$-inch, which rendered their comparisons between the parallel and tapering sides worthless. The efficiency of the latter type was low, as would reasonably be expected.

In /

1. "The Design and Testing of Centrifugal Fans", by H. Heenan and W. Gilbert, Min. Procs. Inst. C.E., 1895-1896, Vol. CXXIII, pages 272, et seq.

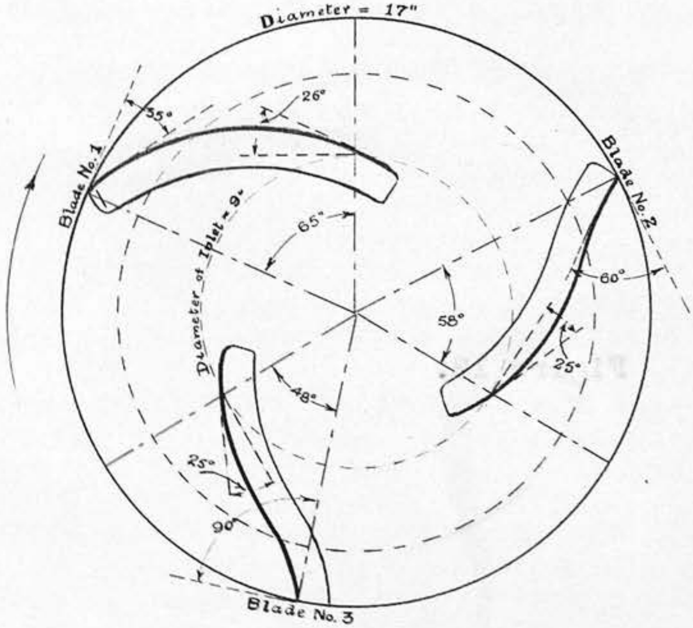

Figure 18. Forms of Blades tested by Heenan a

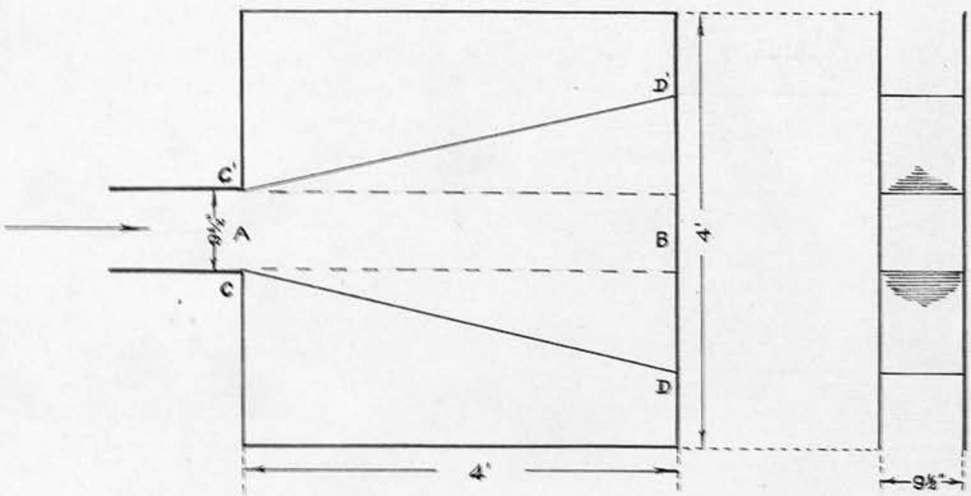

Figure 19. Experimental Evasée used by Heenan
Gilbert.

In their colliery experiment with a 7-feet diameter and 2-feet wide Heenan fan, driven by a horizontal, non-condensing engine a mechanical efficien:cy of 70 per cent is claimed. (Brake horse-power was the denominator). Since, however, a side gauge was used to measure the depression in the fan drift, the figure given exaggerated the efficiency.

Heenan and Gilbert came to the conclusion that a fan with a few simple vanes gives the best results, provided the shape of the blade and the dimensions of the casing are suitably designed. In their opinion, fans of more complex design have too high an internal resistance. (The Heenan fan has from six to eight blades, and as their tests were concerned with this type, their conclusion as to number of blades was unsupported by experimental evidence).

They also tested the efficiency of expanding chimneys. As the correct design of the évasée was one of our chief researches, it is necessary to describe rather fully the work of Heenan and Gilbert in this connection, moreso, since beyond indirect work by Peclet[1] in 1861, we could trace no other efforts of a similar character. Their apparatus is shown in Figure 19, where CD and C'D' are two hinged flaps which could be adjusted to any desired divergent angle between the two parallel side plates. The dynamic and static heads were measured at A (the inlet) and the velocities (at A) and B (the outlet) thereby determined. From the difference in the squares of these respective velocities, the static head at A was calculated. The ratio of the calculated and the measured static heads at A represented the efficiency of the /

1. "Trait de la Chaleur", 3rd edition, 1861; Book III, page 351, et seq.

the évasée for any particular angle. In their own words:-

> " The conclusion to be derived from the
> "tests of expanding chimneys is that in
> "designing a fan chimney the angle on each side
> "may be as much as 15° without loss of efficiency,
> "but means may have to be taken to give a
> "uniform discharge over the outlet area."

The maximum efficiency which they obtained with the angle mentioned was only 47 per cent. Whether, their conclusion has had any influence in the design of évasées erected since 1895 is conjectural. From the particulars of évasées of colliery fans which the writer has tabulated in Part IV, it will be observed that there exists no standard design in the installation of this important fan adjutage.

In general, the methods, results and conclusion of Heenan and Gilbert, in regard to the expanding chimney, will be found to be in marked contrast to those subsequently herein set forth.

The Advent of the Multivane, Drum Type of Ventilator.

While we cannot trace any further published researches of the character of those just described, a distinct advance was made in design when the Sirocco type of ventilator was introduced in 1898.[1] Instead of the few simple vanes advocated by Heenan and Gilbert, the original Sirocco had 80 blades. The number of blades was afterwards reduced to 64 as the result of tests carried out by the inventor. Although the late Sir Samuel C. Davidson - the inventor of the Sirocco fan - spent considerable time in litigation defending patent rights throughout the world, he nevertheless continued researches with the object of improving his ventilator, up to the time of his decease. A scrutiny of the British Patent Specifications affords convincing /

1. British Patent Specification No. 4609/1898. S.C. Davidson, Belfast.

convincing proof of his activities in this direction; between 1912 and 1915 (inclusive) Davidson was granted the rights of seven patents all relating to his fan. Howdver, none of these patents has been developed commercially. We shall have occasion to again refer to Davidson's work when recording the results of our own research.

The salient features of the Sirocco and its contemporaries have already been discussed (See pages 18 and 19).

The remaining works, relating to the centrifugal ventilator, which we now review are more concerned with the theory and practice of the machine than with its design.

Penman's Work.

As stated at the outset, the credit of the present move by the Institute of Mining Engineers towards the simplification of the theory of mine ventilation is chiefly due to Mr. D. Penman who, in the paper mentioned (page 4), ably advocated the adoption of a direct method of assessing mine ventilative resistances, and the abandonment of Murgue's indirect methods of the "equivalent orifice" and "orifice of passage". Penman's suggestion was based on Ohm's law, when applied to a dynamo, thus

$$E = I\ (R + r)$$

where E is the electromotive force in volts, I, the current flowing in the circuit, in amperes, R, the external, and r the ~~ex~~ internal resistances in ohms respectively. The analogous equation in ventilation he proposed was

$$G = Q^2\ (R + r)^1,$$

where /

1. Note: To preserve uniformity in the symbols used, we have altered Penman's symbols to conform with those we have already employed.

where G is the total head produced by the fan, Q, the quantity flowing, R and r the resistances of the mine and fan respectively (for units of measurement see footnote, page 52). The above equation can be broken up, as in the case of the dynamo, so as to express the individual resistances of the mine and fan, thus:-

(1) $Q^2 = \frac{h}{R}$, or $h = RQ^2$

where h is the effective head, and

(2) $Q^2 = \frac{h_o}{r}$ or $h_o = rQ^2$

where h_o is the head absorbed by the resistance and imperfections in design of the fan. As Parker neatly expressed it, R in the above formula, "is the integral of such quantities as K_s/a^3 (of the well-known Atkinson formula) for very numerous small portions of the airway"[1]. The connection between Murgue's methods of inverse measurement of resistance and those proposed by Penman is:-

(1) Equivalent Orifice of Mine $= \frac{c}{\sqrt{R}}$.

(2) Orifice of Passage of Fan $= \frac{c}{\sqrt{r}}$.

where c is a coefficient determined by the units used.

The actual index of Q is, however, a subject for considerable practical investigation. Dr. Penman, in collaboration with his brother, Mr. J. S. Penman, carried out a test at Wellesley Colliery, Fifeshire, and ascertained the relationship between the quantity and pressure for the whole of that colliery under the then existing conditions.[2] Their equation was

$$h = RQ^{1.8}$$

It has been suggested by various authorities that a more correct relationship between pressure and volume /

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1. "The Theory of Ventilation", by Prof. Douglas Hay, <u>Trans</u>. <u>Inst</u>. <u>Min</u>.<u>Eng</u>., 1923-1924, Vol. LXVII; Mr. J. Parker in discussion, page 296.
2. Messrs Penman, op. cit., page 160.

volume for mining work would be

$$h = AQ^2 + BQ$$

where A and B are constants the value of which would be dependent on the mine conditions.[1]

In his original paper, Penman proceeded to demonstrate the utility of the direct method of resis:tance measurement when applied to the solution of problems related to the running of mine fans in combination. Of recent years, problems of this character have frequently arisen where it has been found necessary to augment the volume in circulation, and usually, when two fans in combination have been tried, the results obtained were considerably below those anticipated.

The Series Combination.

Considering the question from the increase in volume point of view, we have, with two identical fans in series,

$$Q = \sqrt{\frac{2G}{R + 2r}} ,$$

the necessary conditions here being that the fans each produce the same total depression, G $(= h + h_e)$, when in combination as when operating separately.

When r is small compared with R, the expression becomes

$$Q = \sqrt{\frac{2G}{R}}$$

This means that the quantity produced by two identical fans in series is $\sqrt{2}$ times the quantity, (Q_1), produced by one. However, to realise this, the total power consumption of the combined fans would be greater than the sum of the powers required when the fans operated singly, since, when in combination, the power would be

$$\sqrt{2}\, Q_2 \times 2G = 2.828\, Q_1 G, \text{ instead of } 2\, Q_1\, G.$$

Consider /

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1. Eg. Hay, Colliery Manager's Pocket Book, 1924; loc. cit page 106.

Consider now what happens when the power supplied to the fans in series is equal to the sum of the powers of the fans when run separately. As already stated (page 58) quantity $\propto \sqrt[3]{H.P.}$, from which it follows that $Q = \sqrt[3]{2}\, Q_1 = 1.26\, Q_1$. In other words, provided the resistances of the identical fans are negligible in comparison with R, and ~~that~~ the fans are run so that the total power supply is equal to the sum of the separately-run power supplies (the fans sharing this equally while in combination), the maximum increase in volume which can be realised is 26 per cent. In practice, however, the fans in series are rarely identical, and their resistances, compared with that of the mine are not negligible; hence, in the series combination, a greatly reduced increase - if increase at all - would be realised than 26 per cent.

The Parallel Combination.

The equivalent resistance of two identical fans[1] in parallel is $\frac{r}{4}$, so that the quantity (Q) circulated by such an arrangement, when both are running at the same speed, is

$$Q = \sqrt{\frac{G}{R + \frac{r}{4}}}$$

whereas, with only one fan operating, the quantity (Q_1) produced, is

$$Q_1 = \sqrt{\frac{G}{R + r}}$$

Comparing these formulae, it is obvious that unless the resistance (r) of each fan is large compared with /

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1. In the electrical case, the equivalent resistance of two identical cells arranged in parallel, is $\left(\frac{r}{2}\right)$, but in ventilation, $r \propto h \propto Q^2$, so that it is necessary to write $\sqrt{r}$ in the latter case, when determining the equivalent resistance for parallel combination.

with R, the increase in quantity resulting from the parallel combination will be very small.

Continuing the electrical analogy, Penman showed the effect on the distribution of the quantity (Q) between two fans in parallel combination, when running at different speeds. A slight decrease in the speed of one would result in the second fan sucking a portion of its air from the external atmosphere, through the fan running at the lower speed. He supported his theoretical deductions by an experiment on two mine ventilators running in parallel.

Briefly, Penman's conclusions are:- The relative resistances of the fans to that of the mine, determine which combination would be the better. In general, where increased quantity is the chief considera:
:tion, the series arrangement would be preferable. In the parallel arrangement, efficient speed regulation was essential. (However, Parker contends that power regulation is of more importance - see under Parker's Work).

The Value of Clive's Work.

Apart from his experimental work in the running of two fans in parallel,[1] in which he obtained somewhat similar results to those subsequently recorded by Penman, Mr. R. Clive has rendered much service towards the general advance in the science of mine ventilation. In his recent work,[2] the varying character of natural ventilation is made manifest. The volume produced by natural causes is solely dependent upon the difference between the average weight of air in the shafts and inclines and upon the resistance of the mine. (Clive, however, did not measure the natural ventilation effects in this way, although he admitted it would have been more accurate to have done so;; he took water-gauge readings at the pit bottom, top of upcast and in the fan drift, in four experiments and therefrom deduced the natural water-gauge for the particular day of his experiments). If the average upcast temperature is the higher, the ventilation will be assisted by the natural agency; if the average downcast is the higher, then the natural ventilation will oppose the work of the fan. Hence, the full effective water-gauge producing ventilation throughout the mine is:-

> the observed water gauge in the fan-drift ±
> the natural ventilation water-gauge;
> the sign of the latter is positive when it aids
> the fan, and negative when it acts adversely.

As Clive points out, the variation in the natural ventilation effect due to changes in the surface temperature may cause a wide variation in the total resistance which the fan is called upon to overcome, quite apart from any change /

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1. "Running Two Fans in Parallel" by R. Clive, Trans. Inst. M. E. 1919-1920.
2. "The True Effect of Natural Ventilation in Deep Mines", by R. Clive; Ibid., 1923-1924, Vol. LXVII, page 273.

change in the mine resistance itself. For instance, at Bentley Colliery, of which Clive is the general manager, it is estimated that the combined resistance under normal working conditions varies from 0.89 Atkinsons in summer to 0.39 Atkinsons in winter, entirely due to the changes in temperature. Since the horse-power in the air required to be dealt with by the fan will vary proportionately, the advantage of having a fan with a good efficiency over a wide range of resistance - i.e., with a flat characteristic curve - is apparent. If economy is to be effected, then there must be a ready means of varying the fan-drift water-gauge according to the season of the year. Where the fan is steam driven this presents no great difficulty, but where a constant-speed motor is the driving agent, the motor, as Parker[1] pointed out in the discussion of Clive's paper, is required to develop a considerably greater power when the natural ventilation is lending its maximum assistance to the mine ventilation, i.e., when it is least required. He (Parker) suggested a three-speed stepped-pulley arrangement fitted to the fan motor as a means of utilising the natural ventilation effects to secure the maximum economy in power consumption. With a large installation, however, the stepped-pulley proposed is scarcely practicable, and a variable-speed motor appears to be the solution in this connection.

It follows, from the full effective water-gauge argument, that the total resistance of the mine and shafts must be determined from that water-gauge, and not the observed (i.e.) in the fan-drift). Clive showed graphically, from the results of his tests that the increase in the total quantity produced by natural ventil:ation decreased as the water-gauge produced by the fan increased, /

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1. Op. Cit., page 299. Parker in discussion.

increased, and that the horse-power in the air due to natural ventilation increased with increasing total quantity.

A very important point which was omitted in Clive's paper and discussion thereon, was the effect of natural ventilation on the determination of the overall efficiency. Generally, this effect is disregarded, which is unjustifiable, since the quantity of air passing through the fan drift is dependent on the full effective water-gauge, and not on the observed. Hence, in cases where the natural ventilation is always in a positive direction, the fan is erroneously credited with the production of the quantity circulated.

Clive's observations also showed the abnormal resistance of the fan-drift inset and the shafts; his figures are, mildly, somewhat astounding. The relative resistances, expressed on a percentage basis, are:-

Fan-drift inset	18.8	per cent.
Shafts	22.1	" "
Mine	59.1	" "

The figures are more surprising when it is realised that the area of the fan drift was 120 sq. feet (it has since been increased), while the shafts, are 1970 feet deep and 20 feet in diameter, and the measurements were taken on a Sunday when the cages were not running.

The true value of Clive's work on natural ventilation is, in our opinion, that it has focussed the attention of the mining engineer upon the magnitude of this factor in mine ventilation. The question of the measurement of the effects of natural ventilation has for long been a thorny one, and with all deference, the complete answer has yet to come. Clive's experimental work was enacted at a colliery where the workings are comparatively flat, so that the difference in the average air temperatures of the two shafts would have /

have afforded more trustworthy results than those recorded. In cases where there are workings extending to the rise and dip - as is most common - the solution of the problem consequently assumes a more complicated character. Clive appeals for further research work, and it is certainly necessary before we know all that we should know on this very important subject. It is possible that a development of the "equivalent resistance" method suggested by Mr. J. Parker, may provide the general solution.[1]

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1. See under "Parker's Work", (next section).

Parker's Work.

To adequately review the work of Mr. J. Parker in connection with the theory and practice of mine ventilators during the past four years would form per se a fitting subject for a thesis. Since December, 1921, he has contributed to the Transactions of the Mining Engineers four valuable papers, all bearing on the mine fan in particular, and mine ventilation in general; further, in the discussion of other kindred papers, his contributions have invariably proved valuable addenda thereto. Together with Professors Briggs, Hay and Penman, Mr. Parker is a member of the group to whom shall belong the credit of having successfully piloted the new unit of resistance measurement through more or less stormy seas. As is apparently inevitable, any proposed departure from deeply rooted practice involves considerable active propaganda work, and it cannot be denied that Mr. Parker has rendered yeoman service in this connection; indeed, his paper, "The Characteristic Curves of Fans"[1], afforded an excellent demonstration of the facility and utility of the direct method in measurement of resistance. In dealing with his work, we shall review it - ever so briefly - collectively.

The Characteristic Curves of Fans.[1]

Parker's main object in this work was to indicate the value of characteristic curves in the predetermination of the output and efficiency of fans, working singly and in parallel, against various resistances. In two subsequent papers[2] he ably illustrated

their /

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1. Trans. Inst. Min.Engs., 1921-2, Vol. LXIII, pages 228-234.
2 (a) "Economy and Efficiency in Ventilation", by J. Parker, Ibid., 1923-4, Vol. LXVI, pp. 14-31.
(b) "The Choice of an Efficient Ventilator for a Mine" by J. Parker, Ibid., 1924-5, Vol. LXVIII, pp.296-309.

their similar applicability to the series combination - in this respect natural ventilation was treated as a particular case of a fan in series with the main ventilator - and also to the most important question, namely, the selection of an efficient ventilator for a mine. The key to such problems lies in the conception of "equivalent resistance". By this term, the value of which is derived from the provisionally accepted law, $p = RQ^2$, is meant the resistance against which the fans are acting when in combination, as distinct from the resistance against which they operated separately. For example, if two fans are paralleled-~~aginst~~ against a constant external resistance, and each fan passes the same volume as it did when running alone, the total volume passed will be doubled while the external water-gauge will be quadrupled. Hence, in parallel combin:ation under such conditions, each fan is passing the same volume as it did when running alone, but at four times the former water-gauge. Or, as Parker expresses it, it is equivalent to putting the fan to work on a mine having a resistance four times as great as that of the actual mine.

He formulated the following two laws relating to equivalent resistance:-

1. When the fans and motors are duplicates, the equivalent resistance against which the fans runn:ing in parallel may be considered to operate is equal to 2^2 times the resistance (R) of the mine; and for other cases is $\underline{n}^2R$, where $\underline{n}$ is the number of times the two fans are together more powerful than the fan running separately. (The condition to be fulfilled is that each fan-motor continues to develop the same power, after the fans are paralleled, as it did when acting alone.

2. If $\underline{n}$ be the number of times that two fans in series are together more powerful than the first running fan, the equivalent resistance ~~aginst~~ against which the fans may be supposed to function is the mine resistance (R) divided by $\underline{n}$.

From these laws it is apparent that the effect of

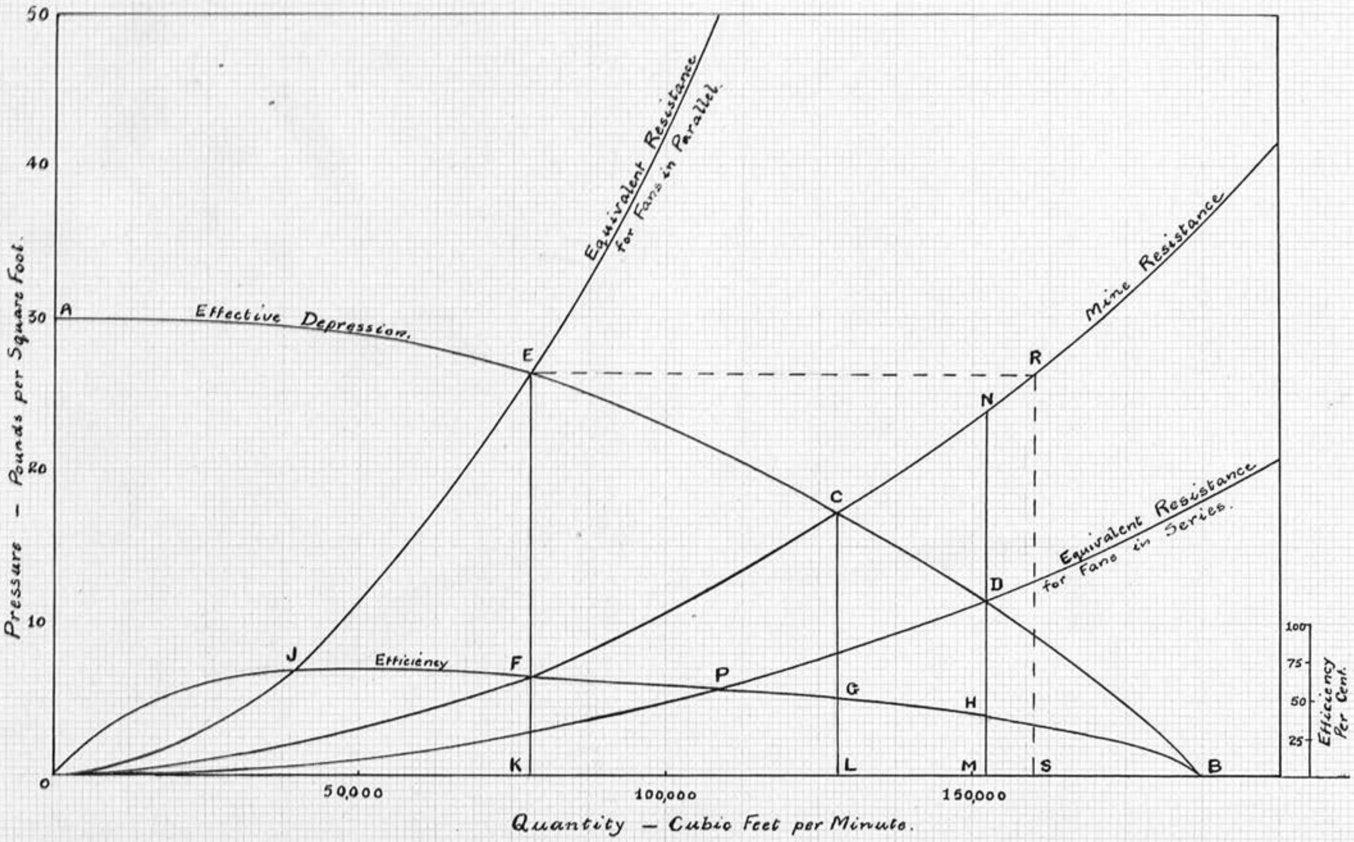
the /

the series combination is to render it easier for the individual fan to pass a given volume; whereas the opposite applies in the case of the parallel arrangement.

In connection with the latter combination, i.e., parallel, Parker makes out a strong case against the dictum enunciated by Penman regarding the necessity of speed regulation. The former holds that the factor of primary importance in the parallel combination is power regulation - that the power input to the fan motors shall either remain constant or shall vary within very small limits. He attempted the experimental verification[1] of his deductions at the same colliery as Penman carried out his experiment in parallel combination, and although he was unable to complete the programme he set out to do, nevertheless, he did obtain evidence supporting his contention.

From Figure 20, which has been taken from one of Parker's papers, we readily see the advantage of the equivalent resistance idea. Suppose AECDB represents the useful depression, OJFGHB the overall efficiency, and OFCR the pressure-volume relations of a fan acting on a mine which has a resistance of 1 Atkinson. The inter:
:section of the resistance and pressure-volume curves gives the ventilating pressure and the volume circulated by the fan when acting on the given resistance. The intercept GL, between the efficiency curve and the volume axis, on the vertical through C, shows that the fan installation is working with an efficiency of 50 per cent. Should it be desired to instal duplicate fans and the question was whether, under the existing conditions, it would be better to arrange the fans in series /

1. "The Operation of Fans in Parallel" by J. Parker, Trans. Inst. Min. Engrs., 1921-22, Vol. LXII, Discussion, Vol. LXIII, page 49.



<u>Figure 20.</u> <u>Equivalent Resistances for Fans in Series and in Parallel: Graphical Determination of Efficiency and Load at which a Fan would operate under various conditions.</u>

series or parallel combination, the answer is readily obtained from the graphs. For duplicate fans in series, the equivalent resistance curve would be OPD; the overall efficiency of each fan is now represented by HM, and equals 40 per cent. The power input has been doubled, but the useful effect has increased in the ratio of 8 : 5 only. Under these conditions the volume circulating has been increased to only 117 per cent. of its former value, as compared with the 126 per cent, increase which the ordinary law connecting volume and power would indicate. The curve OJE represents the equivalent resistance under a parallel combination of duplicate fans; the overall efficiency (FK) for each fan has increased to 65 per cent. in this case, which means that the volume has been augmented to 137.5 per cent. of that circulated by one fan when acting alone.

In his paper "Economy and Efficiency in Ventilation", he describes a "rapid" method wherein the full characteristic curves of a fan may be approximately determined and with very little trouble, (loc. cit., page 15). By this method, the fan is run at a constant speed on two different external resistances, and two measurements only, under each condition, are necessary, namely, quantity and depression.

The Selection of a Fan.

Hitherto, it has been invariably considered that the basic factor upon which the choice of a fan depended was the equivalent orifice of the mine. Parker, in a very able way, shows that the selection of an efficient ventilator is neither dependent on the equivalent orifice, orifice of passage, nor the shape of the blades, but upon two essentials, namely, (1) a consideration of the characteristics of the fan and (2) the limiting values of the external resistance against which the fan is likely /

likely to operate. To provide some degree of regulat:
:ion to efficiently meet the variations in the external resistance, it is obvious that a fan possessed of a flat-topped efficiency curve is of first importance, despite the fact that such a fan will have a relatively high internal resistance. The principle cause of the general low efficiencies in connection with mine ventilator installations is that the fans have been designed for particular "equivalent orifices" which may not have been - or may never be - realised. Once the efficiency curve begins to drop, it does so rapidly, so that should fans be operating under conditions widely different from those for which they were intended, - and unfortunately, this is too often the case - low efficiencies are almost inevitable. It is doubtless that there are many instances of fans installed at some of our large collieries under conditions which, after careful investigation, would indicate that it would be true economy to replace them. Where the annual cost of ventilation for a single mine approaches £10,000, the question of fan efficiency assumes an economic significance. Again, while it must be admitted that, in opening a new mine, the choice of ~~fa~~ a fan to render a lifetime of efficient service is a somewhat hazardous undertaking; nevertheless, now that the real value of the characteris:
:tics of fans is better understood - for which Parker must receive considerable credit - the risks have been greatly reduced.

As a direct outcome from the discussion on a paper[1] by Professor Briggs and the writer, Parker submitted a well-thought out treatise dealing with the selection of an efficient ventilator. Therein, he comprehensively /

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1. "Experiments on the Distribution of Air in Centrifugal Fans", Trans. Inst. M.E., 1923-24, Vol. LXVII, page 8

comprehensively indicates how the component parts of the mine resistance can be estimated after its general lay-out has been settled; the probable variations in resistance due to development of the workings, and also due to seasonal changes are carefully analysed. Citing Clive's figures, Parker contends that the extreme fluc: :tuations of the effective resistance due to natural causes, may, in deep mines, be much greater than any probable changes in the underground development are likely to produce. His method of obtaining the "equivalent resistance" due to the joint action of a fan and a natural ventilating pressure by plotting the latter below the normal volume-axis of the characteristic curves, and then dropping the mine resistance curve so that it commences from the extremity of the extended pressure ordinate, is very ingenious and has possibilities of further development. It rests on an assumption that the natural ventilating pressure is independent of the volume circulating. Provided the fan has been selected to cope with the normal resistance, the effect of the seasonal changes will be to lower the efficiency most when it is assisting in the greatest degree (i.e., in winter), so that the speed of the fan can be reduced.

To illustrate the trend of Parker's arguments on the choice of a fan, we have selected Figure 21 from his paper on the subject. Suppose the minimum desired overall efficiency of the plant be 60 per cent, and that a fan having the characteristics shown in the figure has been offered. A horizontal line drawn through the 60 per cent. efficiency ordinate cuts the efficiency curve at two points, A and B, which points must necessarily be the extreme limits of the desired efficiency range. Drawing verticals, EAC and FBD through A and B respectively, we determine the points, C and D, on the pressure-volume characteristic, through which /

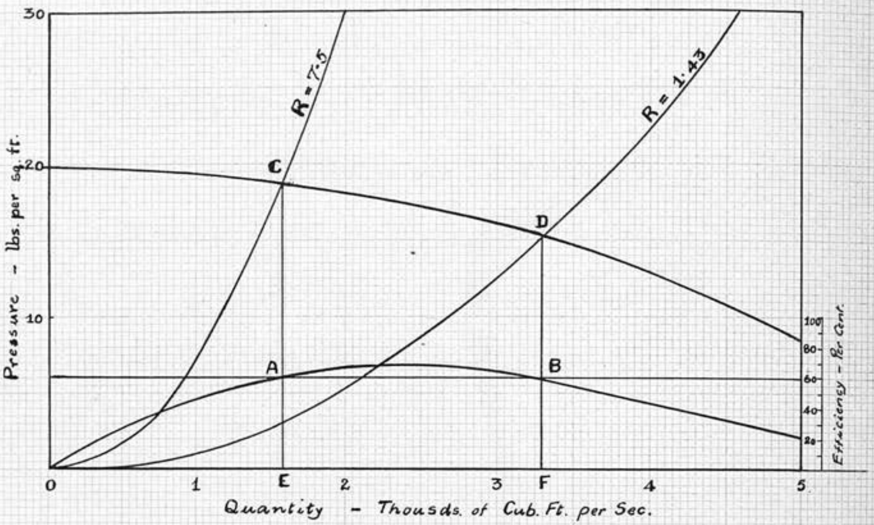

Figure 21. Showing the Characteristic Curves of a Fan and how to determine the Limiting Values of the Resistance within which the fan will give an Efficiency of not less than 60 per cent. The values for this case show that R must not exceed 7.5 Atkinsons and must not be less than 1.43 Atkinsons.

which the curves of limiting resistance must pass. The respective numerical values of these resistances are equal to CE / OE^2 and DF / OF^2 units. If the estimat:
:ed value of the normal resistance of the mine, and the value of such variations of it as may arise, fall between the limits derived from the characteristic curves of the particular fan offered, then such a fan will fulfil the required duty with the desired minimum efficiency. On the other hand, should either or both of the estimated limits of the equivalent resistance of the mine fall outside those derived from the characteristic curves, then evidently an effort must be made to obtain a fan possessed of characteristics which suit the estimated conditions.

In a case where the efficiency of a mine ventilator installation has fallen much below the considered economic limit, due to a large reduction in the external resistance, a remedy suggested by Parker is another fan in parallel, which could be made to boost up the efficiency as near the maximum as desired. Similar:
:ly, should the external resistance have become too great to be efficiently dealt with by the existing fan, the installation of another fan in series is suggested. These suggestions provide alternatives to the replacement of the existing fan.

The extreme value of characteristic curves has been well expounded by Parker, and those responsible for the maintenance of existing ventilators, as apart from those who have to decide upon the installation of new plants would materially benefit from a study of his work. Prospective purchasers of new installations should insist on the guaranteed characteristic curves of the fan provisionally selected being supplied; the ultimate choice should not then be made until satisfactory results, based upon an analysis such as has been briefly outlined, seem /

seem assured.

In "Economy and Efficiency in Ventilation", he suggestively deals with a variety of subjects such as the importance of the correct construction of the évasée, the position of the underground booster fan, fan drives, surface leakage, and excessive resistance of splits. With regard to the latter, i.e., excessive resistance, he reminds the practical mining engineer about an important matter which is too often lost sight of, namely, that the ventilating pressure, and therefore the power required to circulate a definite volume of air through an airway, varies inversely as the fifth power of corresponding sides or diameters, with constant cross-section. This fundamental fact is perhaps made more significant by a practical expression. For example, suppose we have two airways - one, 4-feet x 4-feet in cross-section, and the other of exactly double these dimensions; it would require 32 times the power to force a given volume through the smaller of these airways as would be needed to pass the same volume through the larger airway. The great importance of maintaining as large airways as is practicable is thus apparent.

The Steart Fan.

We have already mentioned the screw-propeller type of fan (pages 9 and 10) and its defects - princip:ally, its inability to produce large ventilating pressures. It would seem, however, from the researches of Mr. F. A. Steart, that this form of ventilator has now to be considered as a serious rival to the centrifugal machine for mine ventilation. In a recent paper, "The Application of Air-Screws to Mine Ventilation"[1], Mr. Steart certainly /

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1. Trans. Inst. Min. Engs., 1924-25, Vol. LXVIII, page 310.

certainly provides strong evidence regarding the efficacy of his invention, - a fan built with modern aircraft propellers, arranged in series on the same shaft.

During the past two years he has conducted practical tests at the Northfield Colliery, Natal, with Curtis aircraft propellers, 8-feet 4-inches in diameter, and of varied pitch. The air-screws were mounted symmetrically and in series on a steel shaft, and spaced 9-inches apart. By making the attachment of the impellers to the boss adjustable, their angle, in relation to the axis of the shaft could be varied, i.e., their pitch was adjustable. Steart used as many as then air-screws in series, and obtained water-gauges as high as 9-inches. His fan was arranged to exhaust air from the mine.

Not only did Steart demonstrate that the old idea regarding the apparent inability of the propeller fan to create pressure was wrong, but he proved convincingly that this form of fan, arranged in series, was capable of circulating large volumes of air against ordinary mine resistances with a satisfactory degree of efficiency. His most recent tests seem to indicate that a definite relationship exists between the number of air-screws, their pitch, and the mine resistance, viz:

(a) the resistance in Atkinsons is almost directly proportional to the number of air-screws;

(b) the pitch is inversely as the square root of the mine resistance in Atkinsons.

If these tentative relationships are confirmed by further investigation, it will become a comparatively simple matter to select a suitable air-screw combination to deal with any particular mine resistance.

Steart also observes that the air passing through the air-screw combination is not given a rotary or spiral motion, but appears to pass from stage to stage /

stage in a purely axial direction; this fact he determined by allowing light pieces of paper to be drawn through the fan. Since each air-screw must pass the same quantity of air, the pressure developed by the combin tion is built up in stages; a series of pressure readings taken between the stages of a six-stage fan demonstrated this. Volumetric ratios as high as 80 per cent. and mechanical efficiencies exceeding 70 per cent. were obtained.

Summarising, the essential features of this form of ventilator appear to be as follows:-

(a) Flexibility - the fan can always be readily adjusted to efficiently cope with variations in the conditions, either by the speed, pitch or the number of screws. Compare this with the installed centrifugal ventilator which has only a speed adjustment.

(b) Cheapness - the difference in cost of such an installation as compared with that for a centrifugal machine must be considerable.

(c) Simplicity - because of its lightness and simple character, the installation of such a ventilator is facilitated.

(d) Reversal - in emergency, simply requires a reversal of rotation: for a permanent reversal, the air-screws on the shaft would be reversed.

As was remarked during the discussion on Steart's paper, the air-screw type would seem to be the fan of the future,[1] A complete experimental ventilator of this form has been installed in the Mining Laboratory of this University and confirmation (or otherwise) of Steart's figures and deductions will soon be forthcoming.

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1. Op. cit., Vol. LXIX, p. 95. Prof. Henry Briggs in discussion.

PART IV.

(A) DISTRIBUTION OF AIR IN PARALLEL-SIDED CENTRIFUGAL FANS.

(B) RE-ENTRY PHENOMENA.

(C) THE ÉVASÉE : ITS DESIGN FOR MAXIMUM EFFICIENCY.

(D) FAN CASINGS.

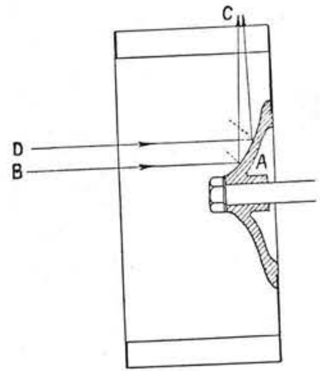

Fig. 22.—Illustrating Rebound of Air from Boss.

PART IV - SECTION A.

DISTRIBUTION OF AIR IN PARALLEL-SIDED CENTRIFUGAL FANS.[1]

(1) Introductory.

Preliminary observations on the passage of light substances, e.g., confetti, through a fan proved that the ingoing air strikes the diaphragm, A, Figure 22, with violence before being discharged through the blades, the impact being the greater the higher the velocity of the air. An air-stream such as B, colliding with the diaphragm, will, unless other forces are acting on it, tend to rebound in the manner indicated. Collisions with other air-streams, such as D, must inevitably cause air-stream B to be deflected nearer the inner end C of the blades. With low inlet velocities, this partial ricochet effect will not be so evident; nevertheless, in fans of the drum type, there will always be a tendency for the air-streams to converge to the inner end of the blades, rather than turn a sharp right angle on entering the fan.

Further observations on the behaviour of a pith ball around the periphery of such a fan when delivering air, clearly indicated the unevenness of discharge across the width of the blades. Indeed, as will be shown, the effective /

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1. The experimental work relating to the manner in which a fan deals with the air it discharges (Sections (A) and (B)) was entirely conducted on fans of the drum type having blades long axially and shallow radially, of which type the Sirocco is the original and most common. It is not to be inferred, however, that the defects in design to which attention is drawn only belong to this type of fan.
Most of the illustrations and diagrams in this Part are taken from papers[2] written in collaboration with Professor Henry Briggs and bearing directly on this work.
2. "Experiments on the Distribution of Air in Centrifugal Fans and on Re-Entry Phenomena", Trans. Inst. M.E., 1923-4 Vol. LXVII, pages 84 - 99.
"An Experimental Study of Fan Evasées", Trans. Inst. M.E., 1924-5, Vol. LXVIII, pages 323-344.

effective part of the blades was limited to approximately two-thirds of their width; while the remaining portion, nearer the inlet, was in some cases not only ineffective, but acting adversely.

So marked were these features, it was decided to attempt the measurement of the distribution of air passing through various fans of the common drum type, under varying conditions of depression and volume.

(2) Method of Measurement.

Before such an investigation could be carried out, a method which would yield reliable results had to be considered. Among the methods tried were:-

(a) The pith ball. This method gave fairly consistent measurements when the fan was running outside the casing, but otherwise was unconvenient. It was used, however, in connection with subsequent experimental work bearing on re-entry phenomena and discussed in the next Section.

(b) The action of acid fumes on moistened litmus paper. Strips of moistened litmus paper were attached to the inner sides of the blades, and air impregnated with acid fumes drawn through the fan. The change in the colour of the litmus was very indefinite.

(c) Confetti. Three blades of a fan were selected, approximately 120° apart, and their surfaces coated with an adhesive substance. Uniformly - sized confetti, 0.2 inch in diameter, and made from the lightest of paper, were introduced, a few at a time, into the air passing to the fan, precautions being taken to ensure their even distribution throughout the ingoing air. Some of the confetti stuck to the treated blades. Dividing the width of the blade into equal divisions and counting the pieces of confetti caught thereon, provided an easy means of determining the proportion of the total air-flow passing through each division.

This latter method gave very concordant results and was the method used throughout the investigations. In all, five fans were examined, four being small fans, and the fifth, a large colliery ventilator. With small fans, numerous tests were carried out while they were delivering various volumes of air against various resistances. Three of them were also tested when running clear of their casing.

Under /

Under each conditions, a test was repeated twelve times, and mean values obtained from which a distribution chart could be drawn. (See Figures 26 to 32). In the test of the colliery ventilator, confetti made from tissue paper, and specially guillotined into half-inch squares, were employed. For obvious reasons, this test was only repeated four times; the fan was running under normal working conditions at the time of test.

(3) Particulars of Fans Tested.

Three of the fans belong to the laboratory of the Mining Department; their dimensions are similar, each being 18-inches in diameter, 6-inches wide and of single-inlet type. One is a Sirocco (see Figure 23) with 64 blades, each $1\frac{1}{2}$-inches deep; the second has 112 straight radial blades of the same depth as the Sirocco; and the third has 56 blades of alternate depths, $3\frac{1}{4}$-inches and $1\frac{5}{8}$-inches respectively, and while the angle at their tips is similar to that of the Sirocco, the blades are otherwise different in shape. Figure 23 shows the general arrangement of the laboratory fan plant; any one of the above three fans can be attached to the same diaphragm and keyed to the motor-shaft A. The gallery, B, is made of sheet steel, approximately 60-feet long, with a normal cross-section, 3-feet by $2\frac{1}{4}$-feet; it is connected to the fan inlet by a converging inlet-duct. The casing is of the overshot type; the side adjacent to the inlet is made of plate glass for observation purposes. The fan speed can be varied over a wide range, as can also the resistance of the gallery.

The fourth fan tested was a small Keith fan, 15-inches in diameter, 6-inches wide, and single inlet, 12" diameter; there are 32 blades, $1\frac{1}{8}$-inches deep at their inlet end and $3\frac{3}{4}$-inches deep at the back end of the fan, their curvature being somewhat similar to that of the Sirocco. This fan is used at the Mine Rescue Station for circulating /

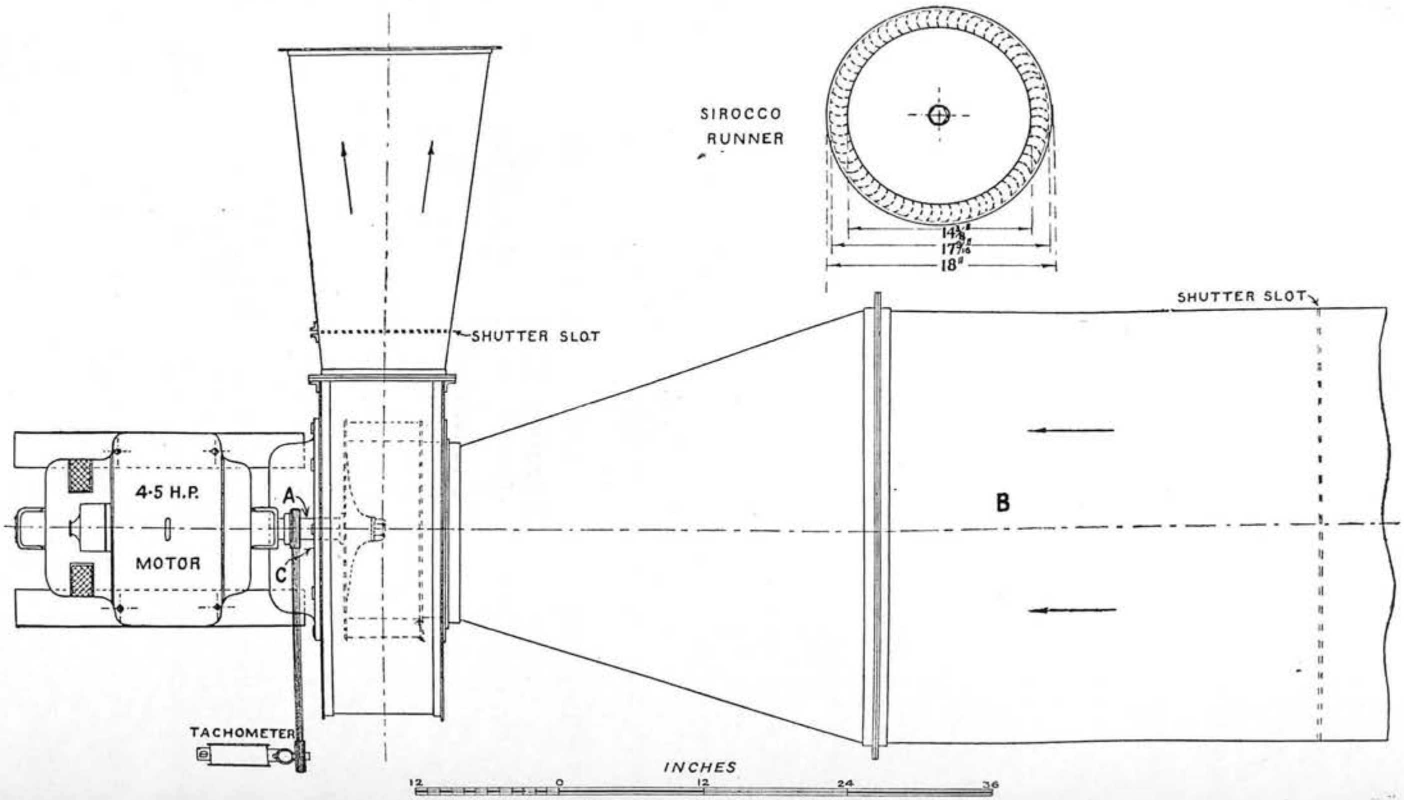


FIG. 23.—ARRANGEMENTS FOR 18-INCH FANS.

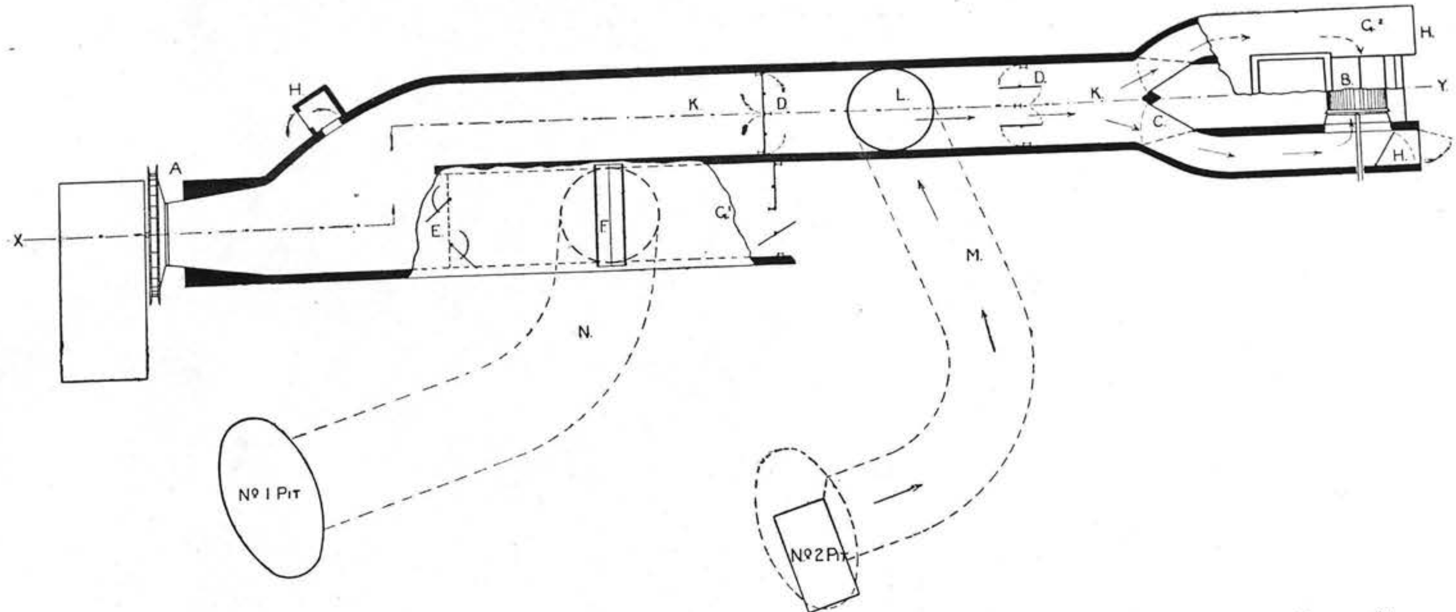
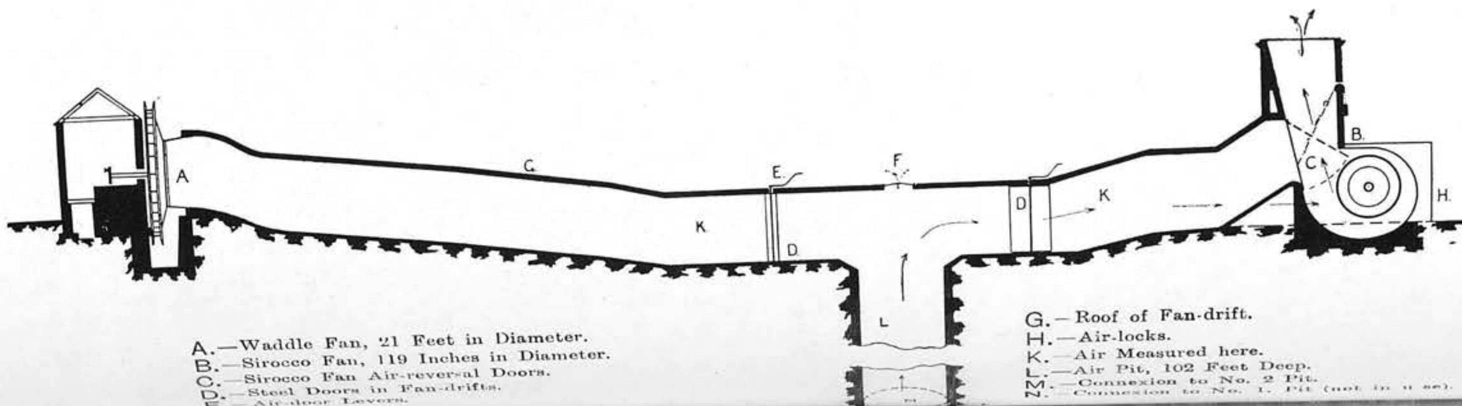


Fig. 24.—Plan showing Arrangement of Duplicate Fans at Wellesley Colliery. Roof of Fan-drifts Removed except at G¹ and G²; Sirocco Double-inlet Fan shown Exhausting. Scale, 32 Feet to 1 Inch. The Reference Letters on Fig. 25 apply also to Fig. 24.



A.—Waddle Fan, 21 Feet in Diameter.
B.—Sirocco Fan, 119 Inches in Diameter.
C.—Sirocco Fan Air-reversal Doors.
D.—Steel Doors in Fan-drifts.
E.—Air-door Levers.
F.—Explosion Doors.

G.—Roof of Fan-drift.
H.—Air-locks.
K.—Air Measured here.
L.—Air Pit, 102 Feet Deep.
M.—Connexion to No. 2 Pit.
N.—Connexion to No. 1. Pit (not in use).

circulating smoke through the practice gallery as required.

The last fan examined was the large double-inlet Sirocco fan installed at Wellesley Colliery, Fifeshire.[1] The duplicate fan plant installed at this Colliery is illustrated in Figures 24 and 25; by the arrangement of doors shown in the figures, either or both of these fans can be connected to the upcast shaft. The Sirocco is 119-inches in diameter and 84-inches wide, while the blades are $7\frac{1}{2}$-inches deep. The inlets of this fan are practically equal to the diameter of the fan itself, the entire depth of the blades being exposed to axially-flowing air at their inlet ends. Only one side of this double-inlet fan was used. Sufficient time was allowed between the stopping and restarting of the fan to ensure that it was running under its normal conditions before carrying out a distribution test.

(4) <u>The Sectional Peripheral Discharge of Air across the Width of a Parallel-sided Centrifugal Fan.</u>

As already stated, numerous distribution tests were made on the various fans examined under widely differing conditions. With the small fans, the resistance to the flow of air in the gallery could be considerably changed without any marked modification in the distribution of the air discharged over the width of the fan blades. For this reason, representative charts (Figures 26-32) only are reproduced. The following table gives the conditions under which each fan was acting when the measurements relating to the respective charts here illustrated were obtained:-

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1. Permission was generously granted for this purpose by Mr. R. Kirkby, general manager, Wemyss Coal Company, Fife. I am also indebted to Mr. J. M. Bell, Chief Surveyor to that Company, for the drawings from which Figures 23 and 24 are here reproduced.

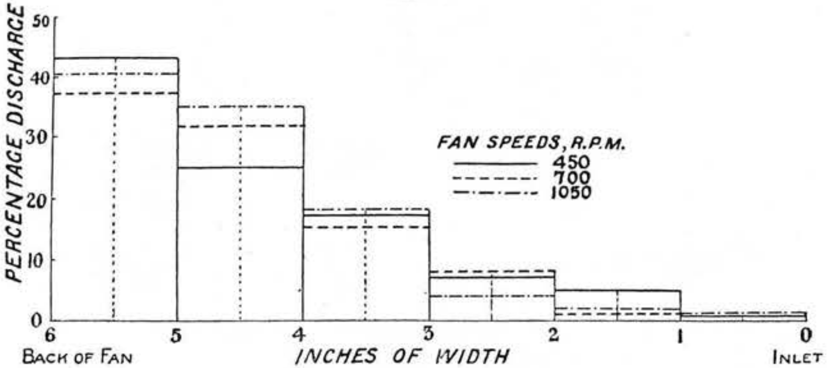

FIG. 26.—DISTRIBUTION OF AIR OVER THE WIDTH: SIROCCO FAN RUNNING IN OPEN.

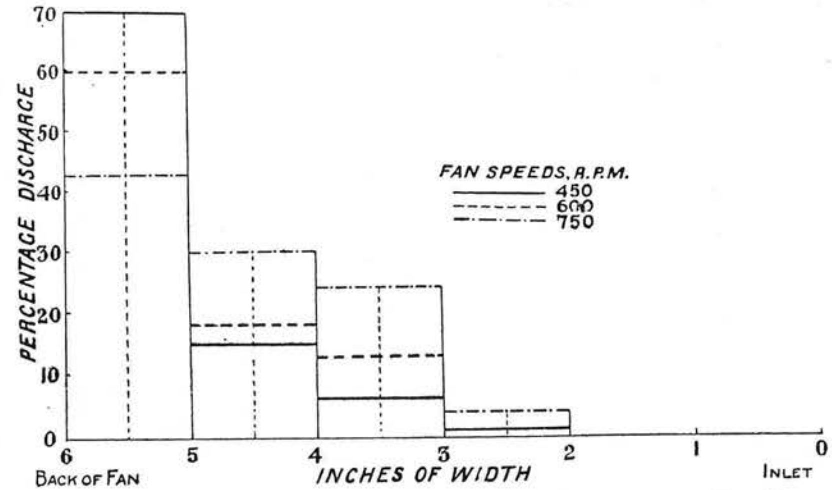

FIG. 28.—DISTRIBUTION OF AIR OVER THE WIDTH: RADIAL-BLADED FAN RUNNING IN OPEN.

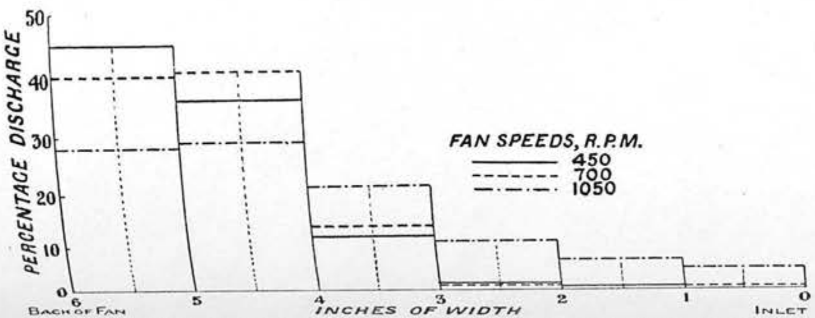

FIG. 27.—DISTRIBUTION OF AIR OVER THE WIDTH: SIROCCO FAN RUNNING IN CASING.

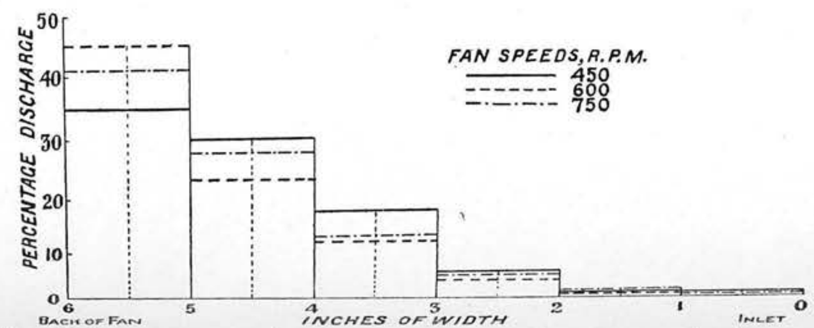

FIG. 29.—DISTRIBUTION OF AIR OVER THE WIDTH: RADIAL-BLADED FAN RUNNING IN CASING.

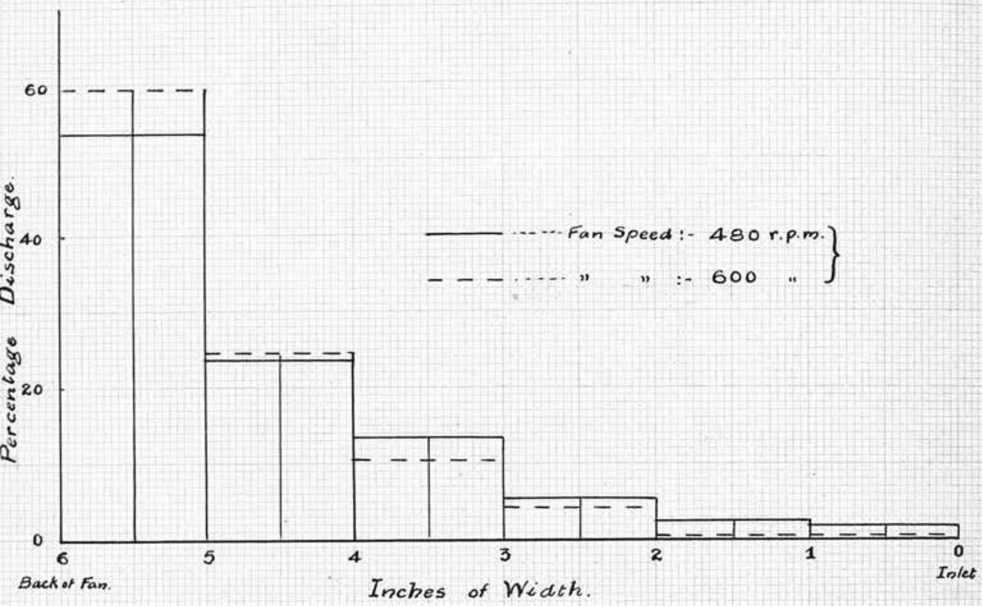

Figure 31. Air-Distribution Chart for Keith Fan.

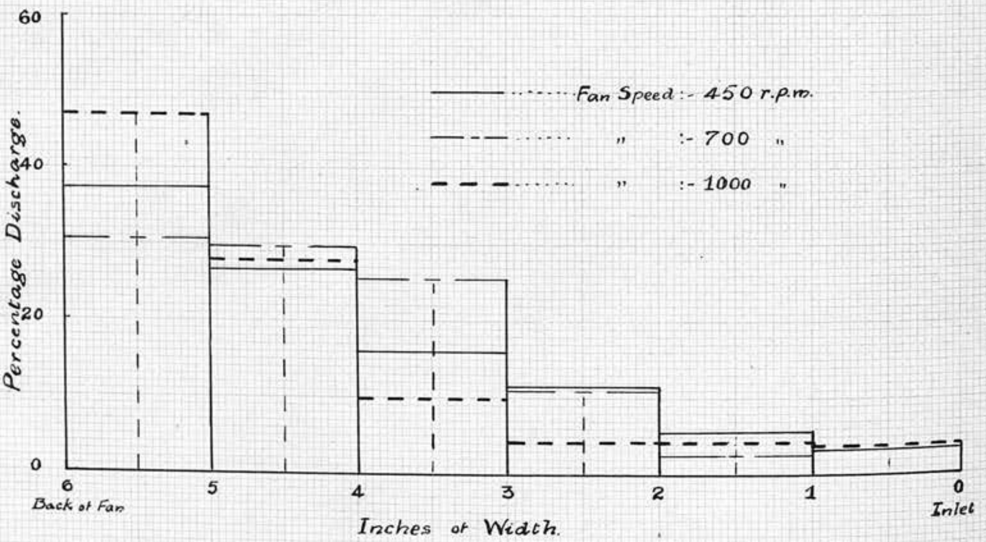

Figure 30. Air-Distribution Chart for Experimental

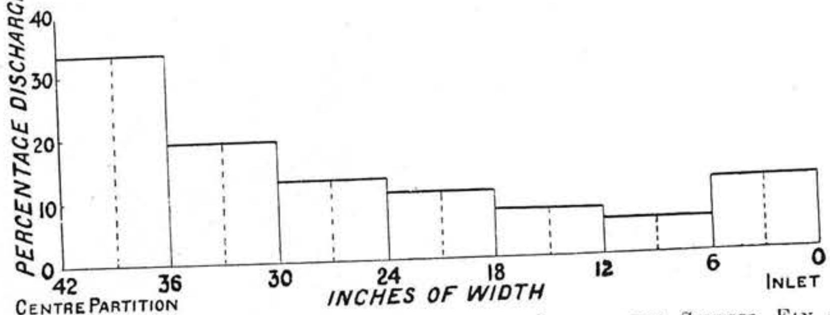

FIG. 32.—AIR-DISTRIBUTION CHART; ONE SIDE OF DOUBLE-INLET SIROCCO FAN AT WELLESLEY COLLIERY.

Figure	Fan.	Fan Speed, r.p.m.	Volume delivered cub. Ft./Min.	Vent. Pres., lbs/sq.ft.	Remarks.
26	18"-Sirocco.	(i) 450 (ii) 700 (iii)1050	- - -	- - -	Fan run:ning in open.
27	do.	(i) 450 (ii) 700 (iii)1050	2234 4218 1957	0.39 0.52 13.5	Fan run:ning in casing.
28	18"-Radial Bladed.	(i) 450 (ii) 600 (iii) 750	- - -	- - -	Fan run:ning in open.
29	do.	(i) 450 (ii) 600 (iii) 750	1336 2248 2634	0.39 0.52 0.55	Fan run:ning in casing.
30	18"-Experi:mental.	(i) 450 (ii) 700 (iii)1000	1073 1740 2600	0.26 0.39 0.65	Fan run:ning in casing.
31	15"-Keith.	(i) 480 (ii) 600	1994 2493	2.60 3.12	Fan run:ning in casing.
32	119"-Sirocco.	154	119654	15.9	Running under normal mine condit:ions.

The charts clearly demonstrate that the inner portion of the blades is passing practically the whole volume of air dealt with by these fans. The proportión of effective work done by each section of a blade along its width is thus markedly ununiform, being greatest at its inner end (adjacent to the diaphragm) and diminishing rapidly towards the inlet end. Indeed, in the next Section, it will be shown that there may be a reversed or negative flow through the inlet ends of the blades. The confetti do not distinguish between normal and reversed direction of flow.

The laboratory Sirocco was designed for a speed of 1050 revolutions per minute; from Figure 27, it is evident that the distribution improves as the fan approaches that speed. In the chart relating to the colliery Sirocco, (Figure 32), there would appear to be a better distribution at the inlet end of the blade width, than occurs in the smaller laboratory fan. This is doubtless /

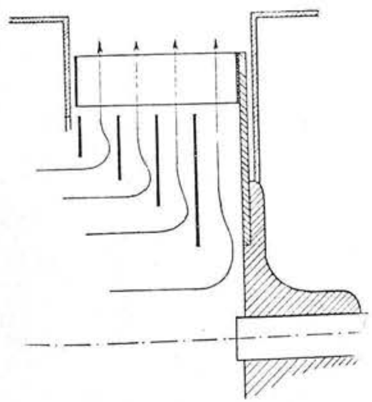

Fig. 33.—Harter's Ring Deflectors.

doubtless due to a difference in the design. As already mentioned, almost the full depth of the larger Sirocco blades is exposed to axially-flowing air, whereas, with the laboratory Sirocco, (and also the other three small fans examined) the inlet ends of the blades are completely hidden by a flat ring to which they are riveted (See Figure 23). Evidently, the former arrangement is more advantageous, although not to the extent indicated by the distribution chart; it is most probable that a small percentage of the confetti caught on the first division of width (i.e., inlet end) would be due to axial and not radial flow.

That the best known fan makers are aware of this uneven distribution is evident. In the Keith fan, as described, the blades are made deeper at their inner than at their inlet ends, presumably in an attempt to equalise the effective work done along the blades. Figure 31, however, would serve to indicate how far that purpose has been achieved. Under British Patent Specification No. 137274/1920, appears a device (Figure 33) due to E. Harter, Paris, which is intended to distribute the air evenly over the vanes of a centrifugal fan. This device may effect its purpose, but at the cost of a reduced efficiency in consequence of increased friction; we cannot trace its application in practice.

(5) Distribution of Air with Reduced Blade-Widths.

Since, in the fans examined, the inlet ends of the blades were shown to be so ineffective, if not adversely effective, further tests were made on the 18 - inch Sirocco to determine whether any improvement could be obtained by a reduction in blade-width. The spaces between the blades were filled up with paper so as to reduce the width to 5, 4, 3, and 2-inches successively. During the distribution tests, measurements were also taken to enable the overall efficiency of the plant to be determined, the gallery conditions being maintained so that a fair comparison of the results /

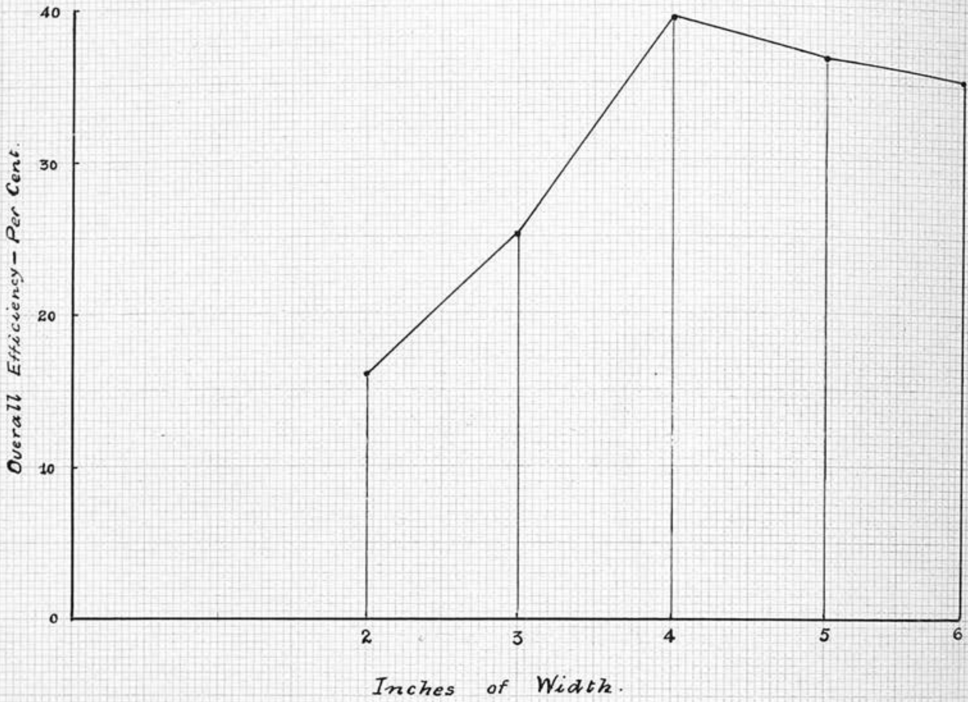

Figure 34. Efficiency of Blade-width - Fan running at Constant Speed.
(Note:- These efficiencies are probably higher than they ought to be, since the dynamic-tube was not set at the position to obtain its mean value).

results obtained could be made from an efficiency basis.

The air distribution charts relating to this series are not reproduced. It is sufficient to record that, as was to be expected, the distribution of the air along the blades became more uniform with decreasing width; in the two concluding tests, (the 3 and 2-inch widths respectively) the distribution was practically uniform.

In Figure 34, a representative graph is given of the efficiencies obtained for the different blade-widths while the fan was running at its maximum speed (i.e.,1050 r.p.m.). The maximum efficiency was obtained when the width of the blade was reduced to 4-inches.

As the work recorded in the next Section is in some respects inseparable from the air distribution experiments, conclusions bearing on the latter work are given at the end of Section (B).

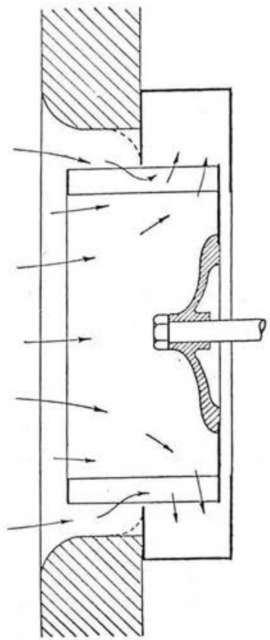

Fig. 36.—An Arrangement of Davidson's.

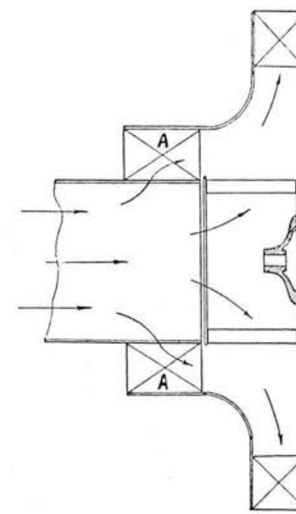

Fig. 37.—An Arrangement of Davidson's.

PART IV - SECTION B.

RE-ENTRY PHENOMENA.

(1) Introductory.

The movement of air by mechanical means is frequently compared to the flow of water created by a pump. However, the wide disparity in the physical properties of these two fluids does not permit the analogy to be carried too far. The extreme mobility of air makes re-entry a much more serious problem in the design of centrifugal fans than in pumps acting on the same principle. Indeed, the re-entry problem may be termed the bete noire of the fan designer.

In the fan and its adjutages, we can point to the following forms of re-entry:-

(a) Axial or Longitudinal Re-entry.

In the previous Section, it has been shown that the inlet ends of the blades were comparatively ineffective. That portions of this blade-width were actually functioning in the reverse direction was demonstrated by the use of a pith ~~bulb~~ ball which revealed the form of re-entry illustrated in Figure 35. This adverse feature could be traced, in varying degree, all round the fan's periphery within the region previously described as ineffective. Thus in parallel-sided wide fans it would seem that the blade-width is partly positive and partly negative in its action.

It is clear that axial re-entry is an effect caused by the unequal distribution of the air in the fan. The inventor of the Sirocco fan was evidently alive to this defect; Figure 36 illustrates an arrangement of his, which bears upon the position of the fan relative to the casing.[1] Another idea of Davidson's is shown in Figure 37.[2] In this case /

1. British Patent No. 1,476, (1912), S.C. Davidson, Belfast.
2. " " " 11,995, (1915), Do.

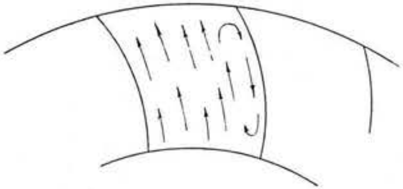

FIG. 39.—RE-ENTRY BETWEEN THE BLADES.

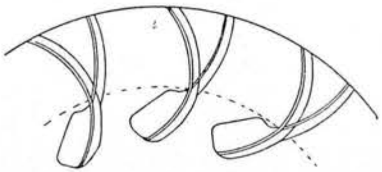

FIG. 41.—CAPELL'S MAIN-AND-TAIL BLADES.

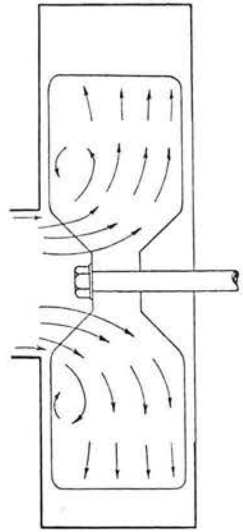

FIG. 40.—LONGITUDINAL RE-ENTRY IN DEEP-BLADED FAN.

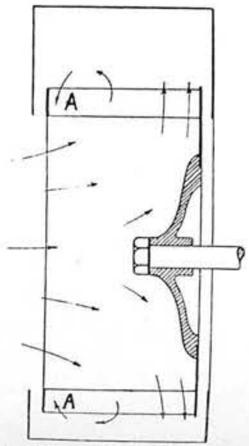

FIG. 35. — ILLUSTRATING LONGITUDINAL

case, an induced flow of air is created by the introduction of an annular inlet A of larger diameter than the fan-runner. The air discharged from the latter into the casing is supplemented by the induced flow through A. This arrange:
:ment is somewhat similar to that of the by-pass B, in the Mortier Fan (Figure 38), to which fan further reference will be made. From the Sirocco firm we gathered that neither of these patents had been developed commercially; it would thus appear that they had not improved the efficiency of the fan.

(b) Re-entry between the Blades.

While we were unable to demonstrate the existence of the form of re-entry depicted in Figures 39 and 40, there is little doubt that it does occur. The type of re-entry depicted in Figure 40 is really both "axial" and "between the blades", since in deep-bladed fans, the former type of re-entry is less likely to make itself evident at the periphery of the fan. The extent of this eddying between the blades is obviously difficult to determine. One of the uses which Professor King (McGill University) suggested for a hot-wire anemometer[1] he designed was for problems of this character; by connecting the anemometer through low-resistance slip rings an analysis could be made of air velocities in the region of rapidly revolving aeroplane propeller blades or between the blades of a centrifugal fan.

Re-entry between the blades will be more pronounced in fans having few blades than in the multivane type. It will also be a defect in parallel-sided fans having an inlet small in relation to the fan-runner, since the vel:
:ocity of the ingoing air will be abruptly checked after entering the fan. It may be further inferred that the extent of this form of re-entry will be inversely propor:
:tional /

1. British Patent No. 18563 (1914) L.V. King.

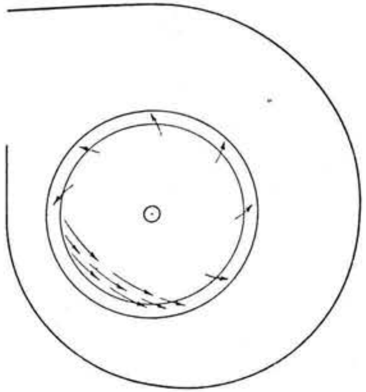

FIG. 42.—CENTRIPETAL RE-ENTRY; FAN-DRIFT CLOSED, ÉVASÉE OPEN.

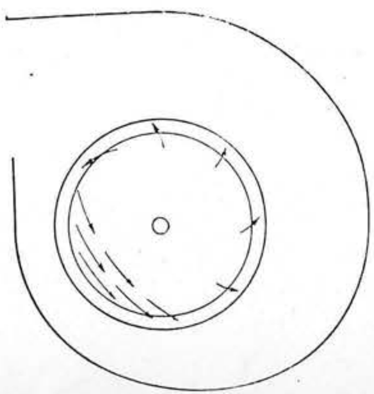

FIG. 43.—CENTRIPETAL RE-ENTRY; FAN-DRIFT CLOSED, ÉVASÉE CLOSED.

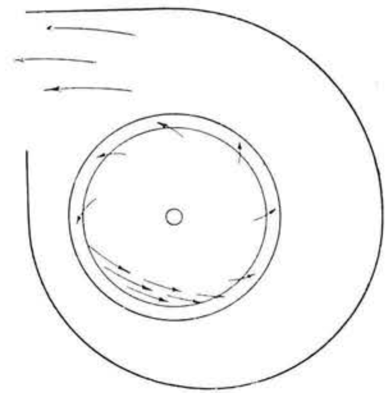

FIG. 44.—CENTRIPETAL RE-ENTRY; FAN-DRIFT FULLY OPEN; FAN DISCHARGING 5,300 CUBIC FEET OF AIR PER MINUTE AT 0·03 INCH WATER-GAUGE.

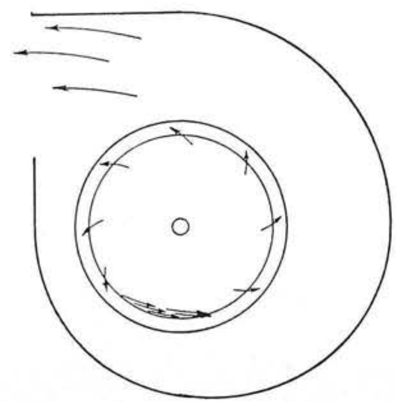

FIG. 45.—CENTRIPETAL RE-ENTRY UNDER CONDITION OF MAXIMUM OVERALL EFFICIENCY; FAN DISCHARGING 3,100 CUBIC FEET OF AIR PER MINUTE AT 2·6 INCHES WATER-GAUGE.

propottional to the volume passing.

The feature of the so-called Ordnance fan is the corrugated blading which is supposed to minimise "blade-slip". Figure 41 illustrates an arrangement introduced by the late Rev. G.D.M. Capell[1], for the seeming purpose of reducing this form of re-entry in his parallel-sided fan; the addition of the "tail-wing" gradually narrows the outlet passage and thereby decreases the re-entrant tendency.

(c) Centripetal Re-entry.

In a discussion on the flow of air through centrifugal machines it would appear wholly irrevelant to suggest the possibility of air flowing from the outside to the inside of the fan, as shown in Figures 42 to 45. These figures are reproduced from sketches made while inside the fan-drift investigating, as far as was possible, the flow of air in the 18"- Sirocco fan; the direction of the air-streams was determined by a pith ball - a morsel of pith secured to a thread, the latter in turn being attached to a stick. The conditions governing each of the four cases illustrated are given with the figures. The centripetal re-entry was most pronounced when both the fan-drift and the évasée were sealed up and the fan was simply churning the air. In Figure 44, the effect is still marked; the fan in this case was passing its maximum volume. When the conditions were such as to allow the fan attaining the maximum overall efficiency, the region of centripetal re-entry was considerably decreased but nevertheless distinctly evident (Figure 45). Because of the tie-rods which act as stays in the large Siroccos, it was not possible to test the Wellesley fan for this effect.

The /

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1. British Patent No. 25468 (1896) G. M. Capell.

The cause of this form of re-entry is principally the casing, the correct design of which does not seem to receive the attention it merits. From the Figures just referred to, (which are not to scale) it will be seen that the beginning of the "volute" is straight, resulting in a restriction of width between the casing and the periphery of the fan. The re-entry shown in Figures 44 and 45 commences at about the end of such restriction. The form of casing is a standard Sirocco design. We return to this feature in Section (D).

The Mortier fan (Figure 38), a French fan, is designed to take in air <u>centripetally</u> and discharge it <u>centrifugally</u>. This diametral-flow type of fan has met with a fair measure of success on the Continent for many years. It would seem specially suited as an underground installation on a straight road, and it is indeed surpris:ing that its merits in this respect have not been recognised in our own country.

(d) <u>Re-entry from the Casing into the Fan-Inlet</u>.

In the discussion of Mr. D. M. Mowat's paper on "Facts and Theories Relating to Fans"[1], Mr. Mark Brand said: "There was a considerable leakage of air from the delivery to the ears of the fan along the clearance between the fan-cheeks and the sides of the building". While we could not trace such a form of re-entry in the laboratory plant, there is no doubt it is a defect which is highly probably in some fan installations, <u>e.g.</u>, fans with backward-trending blades. In fans of the Sirocco type, the "static" pressure possessed by the air immediately on emerging from the blades is lower than the static pressure inside the fan, because of the large proportion of kinetic energy contained in/

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1. Trans. Inst. M.E., 1912-1913, Vol. XLIV, page 248.

in the effluent air. Hence in such a case, there is less tendency for the form of re-entry commented on by Brand, to occur; if leakage does take place, it will be from inside the fan to the casing. It was observed that if a hole was made in the casing, say at C, Figure 23, leakage was inwards and not outwards. The arrangement illustrated in Figure 37 is in some measure intended to take advantage of this feature in fans with forward-trending blades.

To eliminate the form of re-entry under discussion, a device is described under British Patent No. 960, of 1914 (Cattaneo, Italy), and consists of "an: :nular flanges or extensions of truncated conical shape arranged on each side of the casing concentrically with the shaft". The distance between the inside edges of these flanges is slightly less than the width of the blades at the periphery, so that they cause a slight convergence of the effluent air.

(e) Re-entry in the Evasée.

This form of re-entry has been frequently commented upon[1]. The following are the sectional discharge velocities at the mouth of an évasée, $5\frac{1}{4}$-feet long, and sides hading at 7°, when attached to the fan plant shown in Figure 23. The velocities were measured by the $2\frac{1}{4}$-inch anemometer already mentioned, and are expressed in feet per minute; the negative sign indicates that the air was re-entering the évasée at that position:-

I

Fan Speed, 600 r.p.m.

-342	-42	564
- 38	-46	634
214	418	898

II

Fan Speed, 1000 r.p.m.

-388	-88	906
- 98	-98	1344
362	748	1458

1. Eg., Mr. Kerr, in the discussion of J. Parker's "Econ: :omy and Efficiency in Ventilation" said that when measuring the quantity of air discharged from the évasée of a new colliery fan he was testing, he found "in two of the squares (the discharge area was divided into four squares) there were positive and in the other two negative velocities". (op. cit., Vol. LXVII, page 12).

These figures demonstrate that little more than half the discharge area was acting positively. Figures 46 and 47 show how such re-entry may occur. Clearly, to attain as nearly as possible uniformity of flow throughout the évasée, there are two essential factors, namely:-

(a) that the air entering the throat of the évasée is flowing uniformly over that cross-section.

(b) that the sides of the évasée do not diverge too rapidly.

The subject of évasée design was our next investigation, and the record of such is given in Section (C).

CONCLUSIONS - SECTIONS (A) AND (B).

1. The width of the present multivane parallel-sided fan is excessive; in single-inlet fans of this type, the ratio of width to diameter should not exceed 1 : 4.

2. Even although the blade-width be so reduced, unless means are provided gradually to guide the ingoing air from the axial to the radial direction (as exemplified in the Rateau), uneven distribution of the air, with consequent axial or longitudinal re-entry and loss in efficiency, is inevitable.

3. As a result of the uneven distribution, the radial velocity, U, is always greater than is determined from the usual formula (Equation (1)). Also, for the same reason, the kinetic energy losses are always higher than they would be were the velocity uniform.

4. Re-entry between the blades is largely dependent on the shape of the passage between them; they should gently converge towards the outlet, especially in fans with backward-trending blades.

5. The correct design of the casing is of great importance; its shape should be such that it accommodates the increasing volume without creating undue compression at any point, and hence reduces the possibility of centripetal re-entry. The addition of a diffuser between the fan-wheel and the casing (as in the Rateau fan), has much to commend it.

6. Re-entry in the évasée can best be prevented by ensuring that the entering air-flow is centric, and that the divergence of the sides of the évasée is not too rapid. The conclusion of Messrs Heenan and Gilbert in this latter connection requires modification.

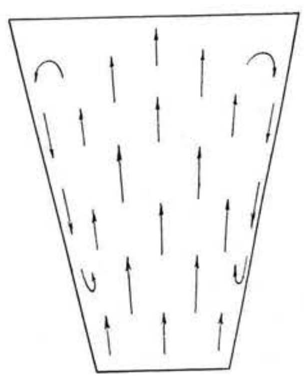

Fig. 46.—Évasée with Centric Stream, illustrating Re-entry.

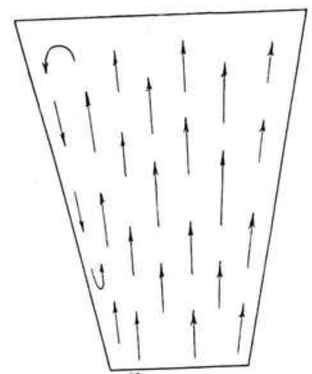

Fig. 47.—Évasée with Eccentric Stream, illustrating Re-entry.

PART IV - SECTION C.

THE ÉVASÉE: ITS DESIGN FOR MAXIMUM EFFICIENCY.

(1) Introductory.

In this experimental study of the flow of air in converging and diverging ducts, our chief purposes were to determine:-

(a) the limiting angles of divergence in an évasée for maximum efficiency;

(b) the economically limiting length of an évasée the sides of which hade at the empirically determined best angle; and

(c) the actual value of the évasée as a means of converting kinetic energy into pressure energy.

Attention has already been directed to the marked occurrence of re-entry in évasées and to the frequency with which such a defect has been commented upon. These factors, together with the very evident absence of any standard form of évasée, promoted our investigation.

The causes of re-entry in évasées were given in the preceding section.

(2) The Function and Principle of the Evasée.

The raison d'etre of the évasée is too well-known to require other than a brief statement of its principle. Air discharged from a fan into an évasée possesses two forms of energy,[1] static and kinetic; due to its velocity at the throat of the évasée, its store of the latter form of energy is large, and unless means are taken to convert this kinetic energy into the static form, considerable loss would result from shock. The function of the évasée, which marks the last stage in the mine ventilating circuit, is to effect this conversion.

The expanding chimney provides one of the best practical examples in the application of the Bernoullian theorem./

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1. That form of energy, termed potential, dependent on the height of a column of air above a certain datum is considered negligible in this case.

theorem. Where p is the amount of static energy recoverable in the évasée, its value is expressed thus:-

$$p = \frac{w}{2g} (V_1^2 - V_2^2),$$

where V_1 and V_2 are the entrance and exit velocities respectively. When the latter velocity is zero, (as it would be with a perfect évasée), then

$$p = \frac{wV_1^2}{2g} \text{..................................} (24).$$

Bernoulli's assumptions of frictionless and stream-line flow are never realised in practice; neither is the velocity at the mouth of the évasée zero. However, equation (24) is useful in assessing the efficiency of the évasée.

A neat way of considering the évasée was suggested to the writer by Professor Briggs. The dynamic water-gauge, as we have previously stated, automatically records the algebraic sum of the "static" and "velocity" heads. Thus, when multiplied by the volume of air passing and a constant, the dynamic-gauge evaluates the algebraic sum of the pressure energy and kinetic energy in the air-current. A perfect évasée can then be defined as one wherein the mean[1] dynamic gauge-reading is constant throughout its length; in other words, the gauge-reading would be zero from throat to mouth in a perfect évasée. In the actual expanding chimney, however, there are energy losses due to skin-friction and eddying flow; the dynamic gauge thus becomes the medium of assessing such losses.

(3) Previous Experiments on Evasées.

While Guibal introduced the évasée we could trace no publication by him relating to its design. At Crachet-Picquery /

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1. Since, because of the varying rates of flow over any cross-section, the dynamic gauge-reading is never constant, the word "mean" is a necessary qualification.

~~2. Wabner, op. cit., page 174.~~

Crachet-Picquery, experiments were carried out by Gille and Franeau[1] on an early Guibal fan, when fitted (1) with an évasée and (2) with no such appendage. They found the mean useful effect in the latter case was 0.2, whereas with the évasée, these workers stated the useful effect to be 0.415. Although these figures cannot be accepted as indicative of the efficacy of an évasée, it is nevertheless true that its influence is more pronounced when the ventilating pressure is low.

An interesting series of experiments on expand::ing ducts were those published by Professor E. Peclet[2]. This able worker, in his experiments on the flow of air was primarily concerned with the determination of the co-efficient of velocity (i.e., the ratio of the actual to the theoretical) relative to various ducts. If friction is neglected, and stream-line motion assumed, then the velocity (V) of a fluid through a duct of constant diameter is calculated from the formula,

$$V = \sqrt{2g}\,H$$

where H is the pressure, in feet of air-column, creating the flow. The frictionless and stream-line flow post::ulated is unattainable in practice, and hence the actual velocity (v), in the case considered (flow through a duct of uniform diameter), is less than V. The ratio $\frac{v}{V}$, the co-efficient of velocity, was Peclet's objective. That this co-efficient can be made greater than unity by attaching a divergent cone to ~~th~~ a duct of constant diameter is common knowledge. Peclet used cones varying from 2° to 53° in their apical angle and found that the maximum value of the co-efficient of velocity was obtained when the angle at the summit of the cone was $6^{\circ}36'$. The value was 1.2; it fell below unity when the apical angle was $10^{\circ}20'$ /

1. Wabner, op. cit., page 174.
2. "Traite de la Chaleur", 3rd edition, 1861, Book III, page 351 et seq.

$10^{\circ}20'$ and when such angle approached 40°, the resulting co-efficient was but slightly different from that obtained when no divergent cone was employed. Peclet observed re-entry with apical angles from $10^{\circ}20'$ upwards; he also noted the necessity for maintaining the axis of the cones coincident with that of the tube.

In his experiments, Peclet's apparatus comprised two tubes, each 1.7-inches long, and 0.43 and 0.3-inch in diameter respectively; the lengths of the conical ducts varied from 8-inches to 10.8-inches. The flow of air was produced by constant pressure on a gasometer, vol::umes measured by displacement, and velocities determined therefrom. The fact that the apical angle which he found to give the maximum co-efficient of velocity close::ly agrees with that which we found best for an évasée, where the apparatus and method employed were so different, is indeed encouraging to those who have perforce to conduct experiments with apparatus of toy-like dimensions as were those used by Peclet.

When reviewing the work of Heenan and Gilbert (Part III) we described at some length their experiments on expanding chimneys. The average évasée efficiencies obtained by them when using angles of hade of 3° (the minimum angle used), 15°, and 21° (the maximum angle used) were 48, 40, and 54.5 per cent. respectively; in a series of tests on one angle, their results varied as much as 52 per cent.[1] Such results are certainly conflict::ing.

(4) Particulars of Apparatus used in the Investigations.

The /

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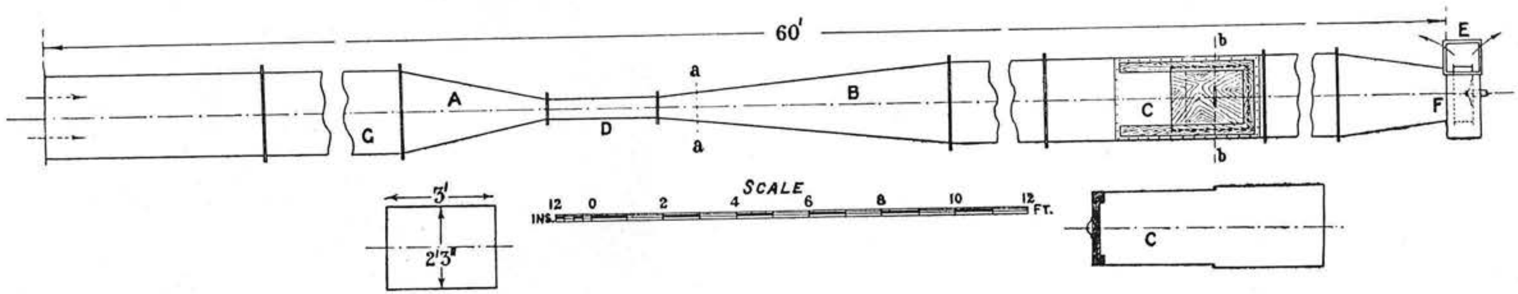
1. Eg. In their tests with an apical angle of 6°, the resulting efficiencies ranged from 15 to 67 per cent. compare this variation with results recorded in Appendix B or C.

The experimental work was chiefly conducted in the laboratory of the Mining Department with the fan plant already illustrated (Figure 23). A more general view of the fan and gallery is given in Figure 48. The converging duct A is 4-feet long, while the diverging duct B is 8-feet long. Their respective cross-sections decrease from 3-feet by $2\frac{1}{4}$-feet (the ordinary section of the fan gallery) to 6-inches by-6 inches, (the cross-section of the wooden rhone, D, 3-feet long, connecting the varying ducts). The ducts A and B can be interchanged, or substituted by other pyramidal ducts. The rhone D can be removed and the convergent-divergent ducts directly connected. In the experiments on the efficiency of convergence, the various ducts used are shown in Figure 49; the concluding test in this series was made on the flat sheet XX in which was cut an orifice, 6-inches by 6-inches, to agree with that of the throat of the divergent portion to which it was attached. Figure 50 gives a sectional plan of the variable divergent arrangement which was fitted inside B (Figure 48). The variable part, A, was 8-feet long and square in cross-section; it was made of $\frac{1}{2}$-inch wood and attached to the wooden shoe C, $7\frac{1}{2}$-inches long, by leather hinges. By altering the sides of A, apical angles ranging from $10^{\circ}40'$ to zero were obtained. Subsequently, two of the sides of A were kept parallel, and only two sides made to vary (as in Heenan and Gilbert's experiments); in this case, apical angles varying from 15° to zero were obtained. Access to the gallery between B and the fan-end was allowed through C (Figure 48), which was closed by a sliding door. The resistance of the gallery could be varied by the insertion of "regulators" through slots in the side of the gallery.

A number of évasées of differing divergence and length were made for attachment to E (Figure 48); short straight /



ORDINARY SECTION OF GALLERY.

SECTION ON LINE $b-b$.

FIG. 48.—GENERAL ARRANGEMENT OF FAN AND GALLERY.

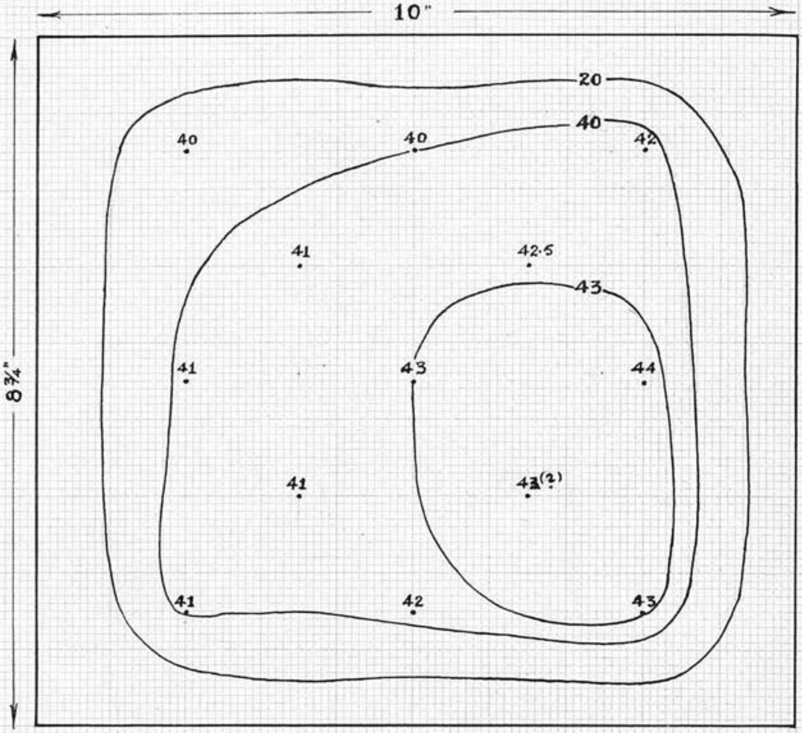


Figure 51.

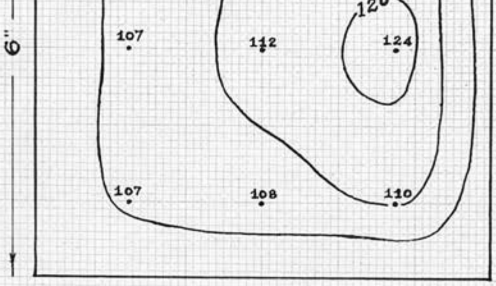


Figure 52. Velocity "Contours" (in feet per second) over Cross-sections of Convergent Ducts.
Quantity: 1,645 cubic feet per minute.

straight lengths of parallel-sided tubes could be interposed between E and these divergent ducts. The necessary measurements of pressure head, and velocity of flow were made with the instruments already described. All important readings of pressure were invariably checked by reversing the limb of the manometer (described on page 41) connected to the gallery. Barometric and hygrometric readings were frequently taken throughout the experiments so that the weight of a cubic foot of air could be determined.

Experiments were also conducted on two colliery évasées of recent installation.

(5) <u>Re-entrant and Turbulent Flow: Its Effects.</u>

Preliminary work concerned detailed explorations over several cross-sections of the ducts A and B, (Figure 48), to determine the extent of the variability in velocity of flow. Four cross-sections were selected in each of these ducts, and the velocity head measured at numerous points over the different areas under various rates of flow. Figures 51 to 54 are representative of these explorations; they illustrate "velocity contours" for the particular areas - <u>i.e.</u>, lines joining points of equal velocity - which have been interpolated between the velocities (in feet per second) calculated from the Pitot tube readings, the positions of the latter being numbered in the figures.

Figures 51 and 52 depict "velocity contours" for two cross-sections in duct B when acting convergently; the apical angles of this duct are 17° and 12° in plan and elevation respectively. The current showed a tendency to cling to one side of the duct more than the other, even although the alignment of the drift was as straight as it could be.

Examining the "contours" shown for the diverging duct - the apical angles in this case being double those given /

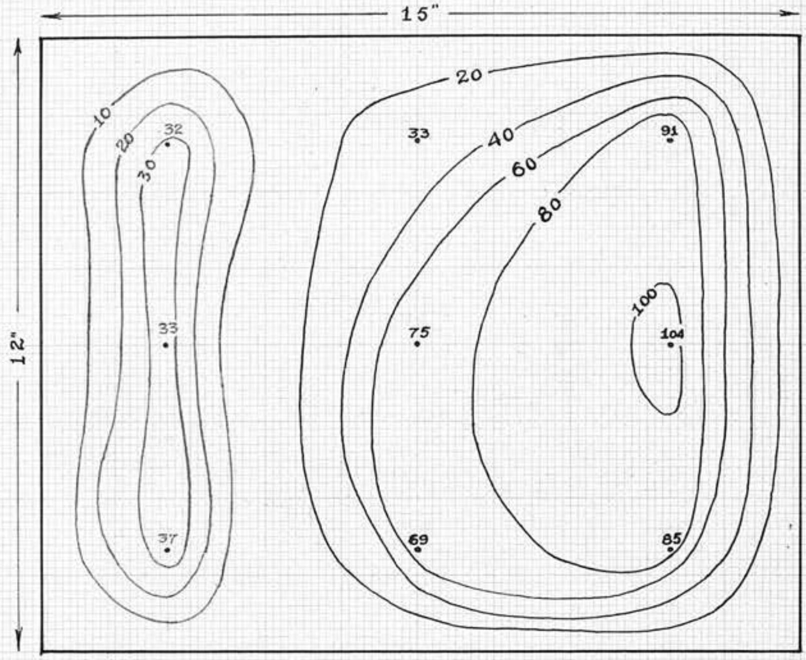


Figure 53. Velocity "Contours" (in feet per second) over Cross-Section of Divergent Duct. "Contours" in red represent reversed flow. Quantity, 2,059 cubic feet per minute.

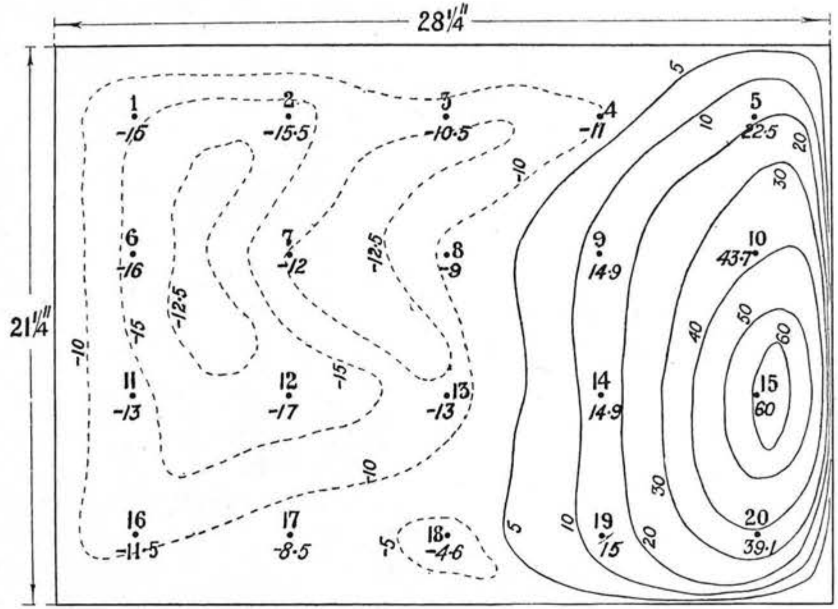


Fig. 54.—Velocity " Contours " over Cross-section of Divergent Duct. The dotted " Contours " represent reversed flow. Quantity, 2,040 cubic feet per minute.

given for B - the bias towards one side of the available area is most pronounced. It was thought that by interposing the rhone D (Figure 48) between the converging and diverging portions, that greater uniformity of flow would be obtained; it made no difference. Another method tried to effect this was the insertion of a thin metal vane, pivoted at the discharge end of D by means of which the air-stream entering the divergent duct could be deflected more to one side than the other; this also produced little or no appreciable improvement. A curious feature which resulted from the use of the rhone D was that on emerging from it, the air could be deflected to either side of the évasée A (Figure 48) by a board (or even one's body); after removal of the deflector, the air stream remained in its new position. A point more worthy of note in connection with the initial experiments with the ducts A and B, was that when B acted divergently (as in Figure 48) an average increase of 27 per cent. in the volume produced by the fan resulted as compared with the volume produced when A was diverging.

The effect of re-entry and turbulence is clearly indicated in Figures 53 and 54. When the observations were taken to enable the latter Figure to ~~the~~ be drawn, 2040-cubic-feet of air were passing per minute. The area of the cross-section is 4.16-square-feet, and thus the average velocity was 8.2-feet per second; actually, the velocity in the normal direction at one point was 60-feet per second, and in the reversed direction, as high as 17-feet per second. Varying the quantity considerably above or below that mentioned did not in any way modify the configuration of the "velocity contours".

(6) Efficiency of Convergence.

In the course of the investigation just described, the remarkable ease with which the efficient transformation of /

of pressure energy into kinetic energy could be effected was apparent. Further tests were made with the various converging ducts shown in Figure 49, the concluding test in this series being made with the flat sheet XX already mentioned.

With the three ducts thus tested, the dynamic gauge-readings were nil throughout their axial lengths when G (Figure 48) was detached. Adding sections of the normal drift to the intake side of the converging ducts, and by the insertion of a regulator, producing turbulent flow before the air current entered them, did not affect the efficiency of convergence; the dynamic gauge-readings along their centre-lines were constant in any one of the varied tests made. With the flat sheet XX, no gauge-reading was registered when a dynamic tube was fixed at the centre of the orifice and various volumes drawn through it.

Thus, no appreciable loss of energy could be detected in the conversion of pressure into kinetic energy. Although "velocity contours" for convergence ducts did not show pure stream-line flow, the observed velocities were not far removed from that state. Among the conclusions arrived at by Professor Osborne Reynolds, from his classical experiments in the two manners of flow,[1] was that a solid converging boundary was one of the conditions tending to maintain stream-line flow.

While we have sufficiently indicated the ease with which pressure energy is changed into kinetic energy without apparent loss, the manner in which the air is dealt with after being subjected to such sudden changes as we effected, must be considered. Had we supported these experiments /

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1. Phil. Trans. Roy. Soc., 1883.

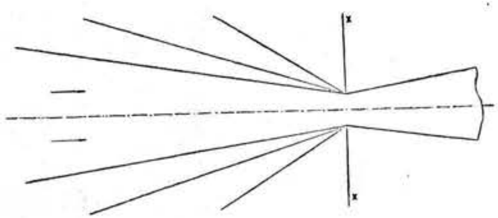


Fig. 49.—Illustrating various Converging Ducts used in the Experiments.

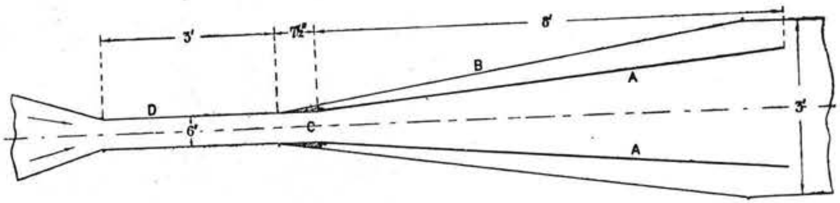


Fig. 50.—Sectional Plan of Variable Divergent Arrangement inside Fan Gallery.

experiments in conversion to kinetic energy by examining the state of affairs existing in a short length of the divergent duct/into which the air from the convergent portions was ejected, we have no doubt that further light on the matter would have been revealed. (See sub-section (9)).

(7) Efficiency of Divergence.

In contrast to the conversion of pressure energy into kinetic energy, to effect an efficient reversal of the operation is extremely difficult. From what has been already said, there is obviously a limit to the rate of divergence if efficiency is to be maintained. To determime this limit was our first objective. The resulting investigations were lengthy and involved several hundreds of pressure and velocity readings.

Because of the undoubted turbulence and pulsation in flow which exists at the discharge orifice of the casing, it was decided not to conduct this experimental work with évasées fitted - as is usual - at that point; the degree of error involved would have been too great. However, tests were subsequently made with a variety of évasées fitted on the discharge. (Sub-section (9)).

It did not in any way alter the problem by placing the expanding ducts at some other part of the system where the prevailing conditions were such as would give greater confidence in the necessary measurements. The principal investigation was consequently carried out with the variable arrangement illustrated in Figure 50 and already described. The method employed was as follows:-

A dynamic tube was placed at a point 18-feet from the mouth of the variable duct, and one-seventh of the width of the drift at that point (measured along the centre line from the side).[1] Another such tube was fixed in a similar position at the discharge end of the rhone D. The /

1. See appendix A.

The difference between the readings of these two gauges, expressed in pounds per square foot, and multiplied by the volume of air passing through the duct per second, was a measure of the energy lost in the évasée due to its imperfections. If the dynamic gauge-reading at the discharge end of the rhone (<u>i.e.</u>, the throat of the diverging duct) had been zero, and had the whole of the kinetic energy possessed by the air on entering the divergent duct been converted into pressure energy by this duct, then the gauge reading 18-feet from the discharge end of it would have been also zero, since the resistance of that length of smooth-sided drift was inappreciable. The dynamic tube on the return side of the évasée was placed 18-feet beyond its mouth so that readings as free as possible from the effects of turbulence and eddying could be obtained.

The volume of air passing during each évasée test over fan speeds ranging from 600 to 1000 revolutions per minute was carefully measured at the inlet end of the drift by means of an anemometer specially calibrated for this investigation. The kinetic energy in the air entering and leaving the évasée could thus be calculated, on ~~the~~ the assumption that the flow was regular. That the flow was, to a certain extent, irregular, over these cross-sections has already been shown; by maintaining the same procedure throughout, however, it was considered that a fair comparison could be made, and if anything, the efficiencies obtained would be lower than their true value. From such data, then, the degree of perfection of any of the ducts examined, could be determined.

In Appendix B is given a representative series of experimental results pertaining to one of the évasées used, the complete reduction of the observations being there set forth; again in Appendix C, two tables are given which summarise the efficiencies obtained for the

two /

two types of évasées examined, namely, (i) uniform divergence on all four sides, and (ii) two sides parallel, and two uniformly divergent.

Figure 55 shows the relation between efficiency and angle of divergence of the 8-foot duct, for the same fan speeds, namely, 900 revolutions per minute; the full line represents the relationship when all four sides expanded equally, and the dotted line relates to that in which two of the sides are parallel. The apical angle which gave the maximum efficiency - almost 80 per cent - was $7^{\circ}10'$ when all four sides diverged, and in the other case, an angle $10^{\circ}42'$ gave the highest efficiency, namely, 70 per cent. While in general, the highest efficiency for each angle was obtained at the highest speed, the difference between the efficiencies for the extreme speeds of any one series did not exceed 6 per cent; more often the difference was much lower.

It will be observed that the dotted curve is comparatively flat between 8° and 14°, and does not drop so rapidly as the curve denoting uniform divergence. A consideration of the rates of expansion of area in the two types of évasées will suffice to explain this. For example, to give a discharge area equal to that of the 8-foot duct hading uniformly at $3^{\circ}35'$, the expanding sides of the parallel-sided duct would each have to diverge at approximately 13.5° for the same length of duct, an angle much too great for high-velocity air to follow closely. The parts of the curves near the origin in Figure 55 are conjectural.

As is well known, the horns of gramophones, loud speakers, and wind - instruments diverge uniformly and moderately at their commencement. The divergence of fourteen different types of trumpets was measured at their necks; the apical angles varied from 4° to 14° and gave an average of 7.8°. While the purpose of these

divergent /

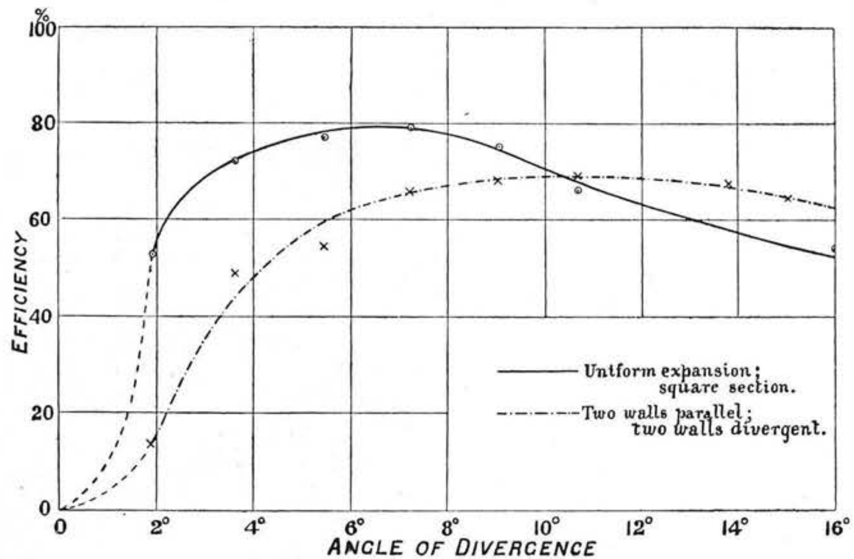


FIG. 55.—RELATION BETWEEN EFFICIENCY AND ANGLE OF DIVERGENT DUCTS.

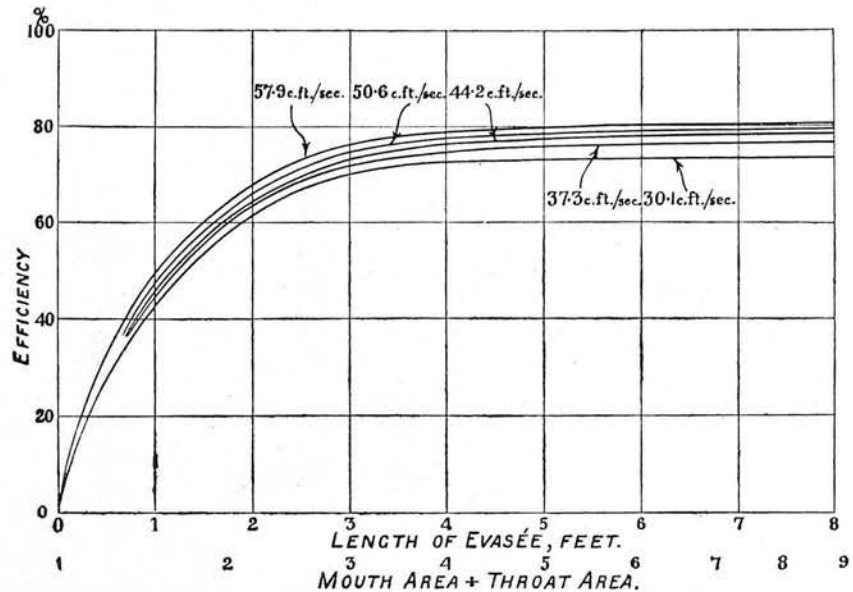


Fig. 56.—Variation of Efficiency with Length; Divergent Duct, 7° 10′ Angle.

divergent funnels is altogether different from that of the colliery fan évasée, the closeness of agreement of the above average angle with that found best for an évasée is nevertheless suggestive.

(8) The Limiting Lengths of Efficient Divergence.

To determine the practical limit in length of évasées, the duct expanding uniformly at 7°10' (apical angle) was selected for exploration. Dynamic gauge-readings were taken at each foot of length, commencing at a point 3-feet from the throat, for five rates of flow. It was not possible, under the conditions, to obtain readings beyond the 3 - foot mark. The gauge-readings were taken at each of the four possible theoretically correct positions, i.e., one-seventh of the width measured along a centre line from the side (see Appendix A).

From the means of these pressure readings, together with the measurements pertaining to rates of flow, cross-sectional areas, and atmospheric conditions, the efficiencies for each foot of the évasée under the five rates of flow were derived. The results of this work are tabulated in Appendix D; they are also graphically represented in Figure 56, by plotting the efficiencies against (i) length of évasée and (ii) the ratio, mouth area: throat area. The curves between the origin and the 3-foot mark were approximately calculated.

It is clear that the greatest loss of energy occurs at the beginning of the évasée, and consequently it is at this portion that the shape of the évasée and the condition of the ingoing air require most consideration. The experimental results depicted in Figure 56, also indicate that no benefit was obtained beyond the 4-foot length of duct, i.e., when the mouth area is to the throat area as 4 is to 1.

(9) Efficiency of Ducts Attached to the Discharge Side of Fan.

Tests were next made with the various ducts shown in Figure 57 to determine their relative influence on the overall efficiency of the laboratory fan plant when the Sirocco fan was run at various speeds against a constant resistance. The volumes circulated (which are printed on the chart) were measured by the anemometer, and the ventilating pressure by a dynamic gauge. The results are set out in the Figure.

Although every reasonable precaution in measurement was taken, the results are not consistent. From what has been shown in Section (A), the air discharged from the fan is highly irregular in its flow; turbulence is very pronounced at the discharge orifice of the casing. It was therefore anticipated that the results obtained at this end of the system would be subject to considerable experimental error.

The maximum overall efficiency was obtained when the arrangement L was employed, the apical angle of the évasée being 7°, which is in agreement with the results recorded under (7) "Efficiency of Divergence". Between that efficiency, however, and the maximum obtained when no évasée was attached (A in Figure 57) there was a difference of only 6.6 per cent. In the arrangement B, a parallel-sided tube, an increased efficiency was obtained compared with the casing discharge orifice results (A).

Further reference will be made to the above results in the next sub-section.

An attempt was made to correct the irregularity of flow existing at the casing discharge orifice by successively attaching the arrangements shown inFigure 58 under the letters B, C, D, E and F. This attempt was /

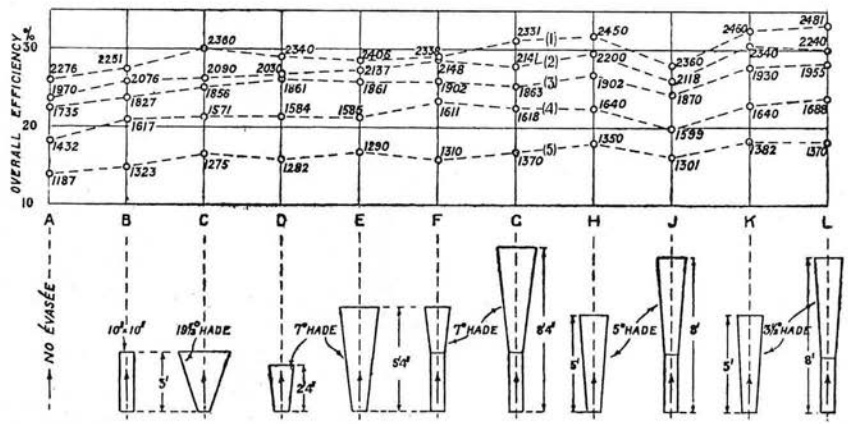


FIG. 57.—THE INFLUENCE OF ÉVASÉES OF VARIOUS SHAPES ON THE OVERALL EFFICIENCY OF A SMALL FAN.

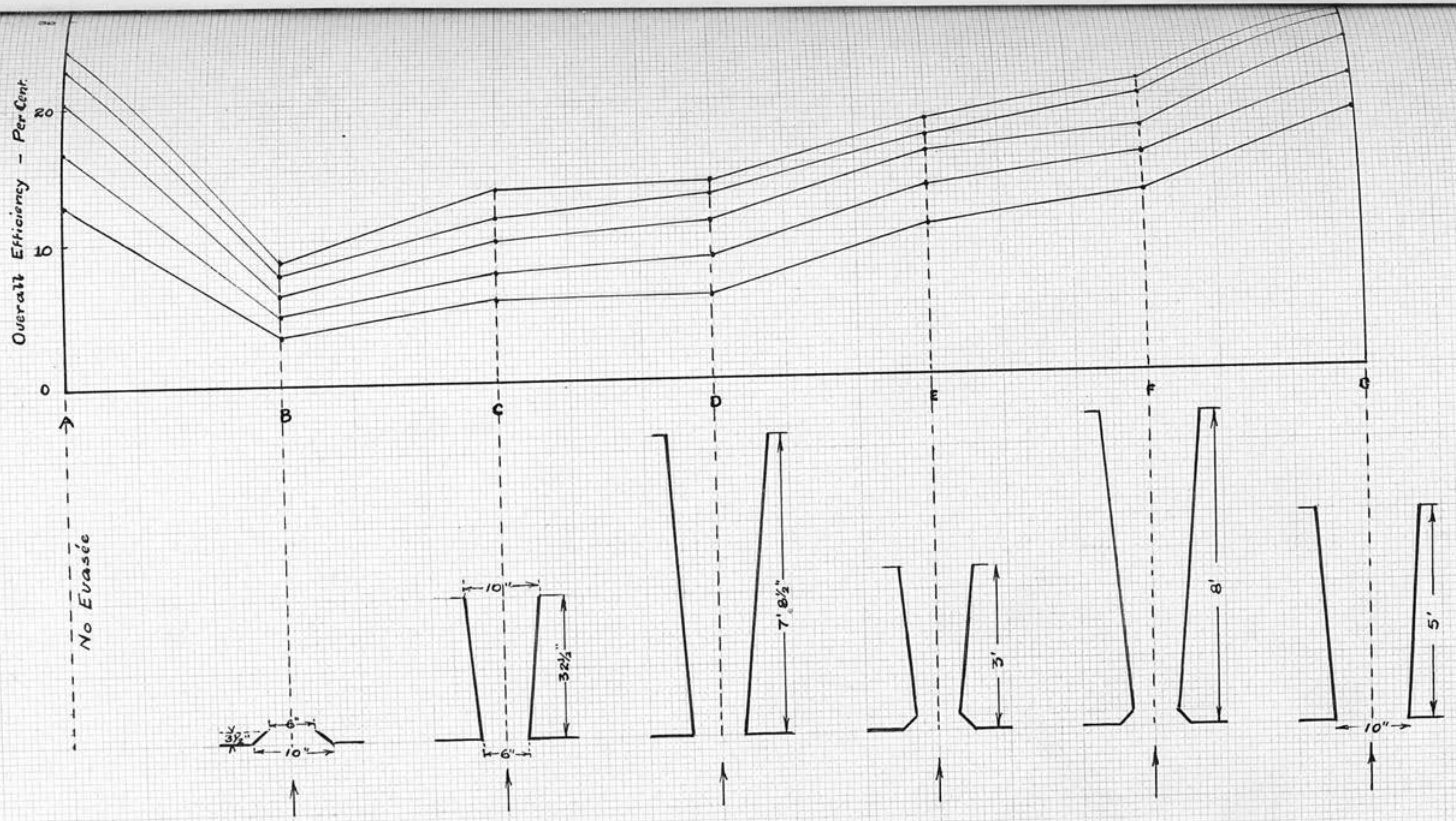


Figure 58. The Influence of Throttling at the Évasée Throat on the Overall Efficiency of the 18-inch Sirocco Fan; all évasées depicted above hade at 7°.

was prompted by the results obtained in the converging duct experiments. The cross-section of the orifice at the casing discharge is 10-inches by 10-inches; arrangements C and D suddenly reduced this area to 6-inches by 6-inches; while arrangements B, E and F reduced it to the latter area in a length of 3½-inches.

The degree of success achieved in this connec: :tion is sufficiently demonstrated in Figure 58; the reduction in the overall efficiency left little doubt regarding the efficacy of the "turbulence correctors". Nevertheless, it is believed that the ideal évasée would be one of vena contracta shape at the throat - the converging portion producing stream-line flow before the air entered the divergent part.

(10) The Actual Value of the Evasée.

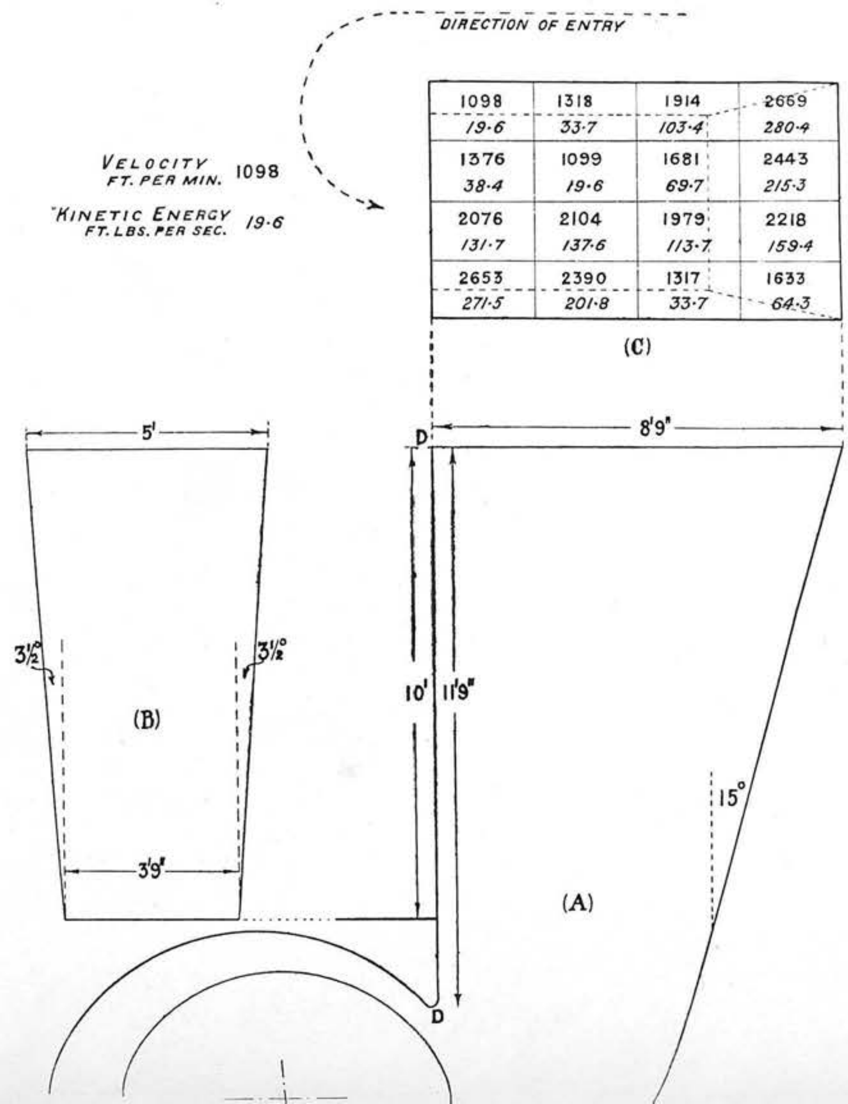
Experiments were conducted with two modern évasées installed at large collieries.[1] Figure 59 gives three views of the expanding chimney of the Keith-Blackman single-inlet fan used at Prestongrange Colliery. This fan is 7-feet in diameter and $2^1/_3$-feet wide; it is designed to deal with 100,000 cubic feet of air per minute against 36 pounds pressure per square-foot. At the time of test, it was circulating 80,370 cubic feet per minute against 17 pounds pressure per square-foot. The month of the évasée was divided into 16 equal divisions by means of string; the velocities, in feet per minute, for each of these squares are given in the Figure (C), the figures in italics /

1. We are again indebted to the management of the Wemyss Coal Coy. for the permission so generously granted in the use of the plant at their Wellesley Colliery.
We are also greatly indebted to Mr. D. Boyd, Manager, Prestongrange Colliery, East Lothian, for the facilities and assistance given during the observations on the évasée at that Colliery.

italics being the divisional amounts of kinetic energy in the air on discharge. While no re-entry could be traced with the anemometer, the variability of outflow is marked. It is interesting to compare this variability of discharge in relation to the direction of inflow to the fan; further support is here obtained to the work recorded in Section (A).

The mean velocity at the évasée discharge was 1873 feet per minute. Using this figure in the calculation of the kinetic energy dissipated, we find such to be 2.82 horse-power; whereas by summing the amounts of kinetic energy divisionally determined, the loss is 3.44 horse-power. Thus, by taking the mean velocity an 18 per cent. error is made in this case. Incidentally, the foregoing gives practical support (if such were necessary) to what has been already said regarding the relative amounts of kinetic energy carried away by an air-current moving (i) with uniform flow, and (ii) with variable flow. The higher the air velocity, the greater will be the disparity in the respective losses of energy. For example, the mean velocity of flow at the throat of the Keith évasée was 4,682 feet per minute. Assuming that the variability in flow at the throat of this Keith évasée was relatively similar to that recorded at the mouth, then the respective calculated amounts of kinetic energy in the air at the throat would be (i) 16.8 horse-power for uniform flow and (ii) 20.5 horse-power for irregular flow.

The total power supplied to this plant was 80.5 horse-power, so that the actual loss due to dissipation of kinetic energy at the mouth of the évasée was 4.3 per cent. Had the height of this chimney been doubled, this loss would have been reduced to 1.25 per cent. The overall efficiency of the plant was 51.2 per /



1098 / 19·6	1318 / 33·7	1914 / 103·4	2669 / 280·4
1376 / 38·4	1099 / 19·6	1681 / 69·7	2443 / 215·3
2076 / 131·7	2104 / 137·6	1979 / 113·7	2218 / 159·4
2653 / 271·5	2390 / 201·8	1317 / 33·7	1633 / 64·3

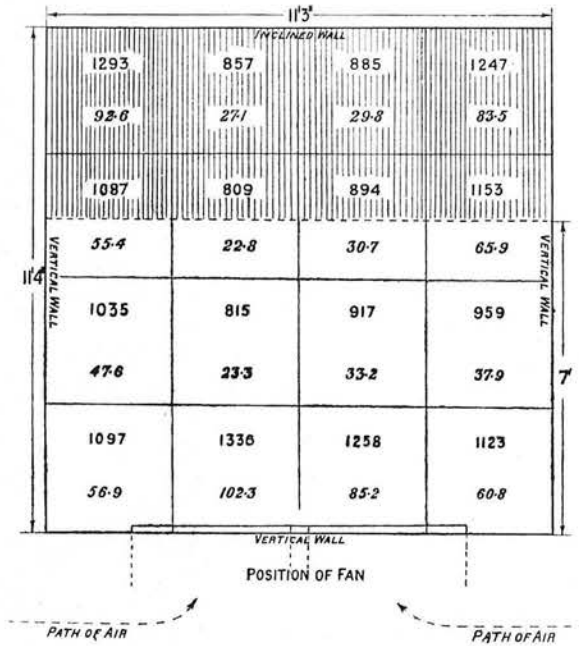


FIG. 60.—PARTICULARS OF DISCHARGE AT MOUTH OF ÉVASÉE OF SIROCCO FAN AT WELLESLEY COLLIERY. THE VELOCITIES (IN FEET PER MINUTE) ARE SHOWN IN ROMAN FIGURES AND THE KINETIC ENERGY (IN FOOT-POUNDS PER SECOND) ARE SHOWN IN ITALIC FIGURES.

per cent.

At Wellesley Colliery, the Sirocco évasée has three vertical sides, while the other side, most remote from the fan, is inclined at 12°; its height is 20 feet. From the results obtained in sub-section (7), the divergence of this chimney would seem reasonable. Figure 60 shows the distribution of velocity at the mouth of this évasée when the fan was delivering 127,600 cubic feet of air per minute against a ventilating pressure of 19 pounds per square foot. In this case, also, no re-entry could be detected with the anemometer. The kinetic energy wasted, calculated from the mean discharge velocity, is only 1.26 horse-power; determined divisionally, this loss is 1.55 horse-power. Making the same assumption as in the case of the Keith évasée, the respective throat values are 3.28 horse-power and 4.1 horse-power. The power supplied to this plant at the time of test was 198 horse-power, the overall efficiency thus being 37.5 per cent.

In this case, then, it would appear that the value of the évasée was very slight indeed, being responsible for the recovery of only 2 per cent. of the total energy supplied, and of which it actually recovers 2.55 horse-power; or otherwise expressed, it loses only 0.78 per cent. of the total power input to the plant, and per se has an efficiency of about 63 per cent. However, as stated elsewhere, this plant is designed to deal with 400,000 cubic feet of air per minute against a pressure of 41.6 pounds per square foot; when the fan is dealing with such a volume, 107 kinetic horse-power will enter the évasée and about 41 will be wasted at the mouth. The value of the correct design of the évasée would therefore be a more potent factor in the saving of energy as the demands on this fan increase.

Another /

Another condition which would have an import:
:ant bearing on the degree of imperfection of these two colliery évasées dealt with, is the reduction of the ventilating pressure while maintaining the same volum:
:etric discharge. In such a case, the total power supply is reduced, ~~the~~ and the percentage of such power which the évasée would be required to recover is relative:
ly increased.

Apart from their evident aberrations, the experiments described in sub-section (9) allow us to probe this question a little more fully. From series (2) of the results shown in Figure 57, an increase of 6.6 per cent. in the overall efficiency of the plant was obtained by using the discharge arrangement L as compared with no discharge duct. The velocities at nine equal divisions of the throat area were measured, and from them, it was ascertained that, under the conditions prevailing when the observations for series (2) were obtained, the kinetic energy at the throat was 0.2 horse-power, or 9.1 per cent. of the total power supplied to the fan-motor. The efficiency of this évasée (which diverged at 7°) was thus ($6.6 \div 9.1 =$) 0.725 or 72.5 per cent. The corresponding efficiency for series (4) was 79.6 per cent. Such figures are comparable with those obtained when the 8-foot evasee, with apical angle 7°10', was used under more favourable conditions (See sub-section (7)).

(11) <u>Typical Colliery Evasées.</u>

Existing fan évasées are either square or rectangular in cross-section. From their varied character in length, angle and elevation, it is apparent that no standard in design exists. In some forms, three sides are vertical, and that most remote from the fan, divergent (as in the Wellesley "Sirocco"); in others, the two side-walls are vertical, while the other /

other two sides are equally inclined; again, the two-side walls may diverge moderately, the wall nearest the fan be vertical, while the remaining wall diverges more rapidly than the side-walls (as in the Prestongrange "Keith); or again, all four sides may diverge equally, or in pairs of different inclination, the side-wall pair hading at a gentler angle than the other pair. Indeed, at one colliery we visited, belonging to the Fife Coal Company, there still exists an old Guibal fan having a square chimney (parallel-sided), 12-feet high; this chimney, however, surmounts the short length of expanding chamber made between the shutter and the casing, (see Figure 4). As we have seen from arrangement (B), Figure 57, the parallel-sided discharge alone has some merit.

Particulars of several typical mine fan évasées have been collected and tabulated (see next page); from the table, the absence of standardisation in design is evident. Except for the two évasées examined experimentally, the particulars tabulated were obtained from drawings of fans. The height or length of the évasées was measured from the throat.[1]

The original angle adopted by Guibal, namely, between 8° and 10° on one side only - that farthest from the fan - is quite sound. It will be observed that the French fans at the bottom of the list adopt a rational design both in regard to angle and length. Figures 61 and 62 illustrate the Galland and the Monnet & Moyne fans respectively; attention is specially directed /

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1. In general, the "throat" is the cross-section at the point where the wall nearest the fan joins the fan casing. With the Guibal, the throat section which is variable in this case because of the shutter (see Fig. 4) was taken at fan-shaft level. In the Rateau, the location of the "throat" was taken to be immediately below the fan-shaft.

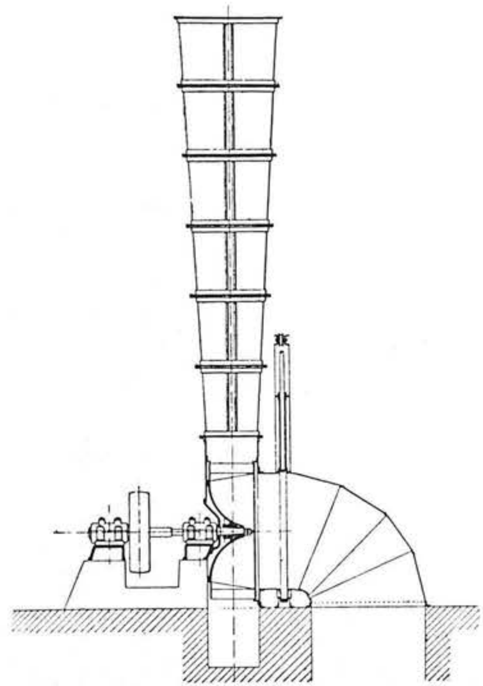

Fig. 61.—Galland Fan.

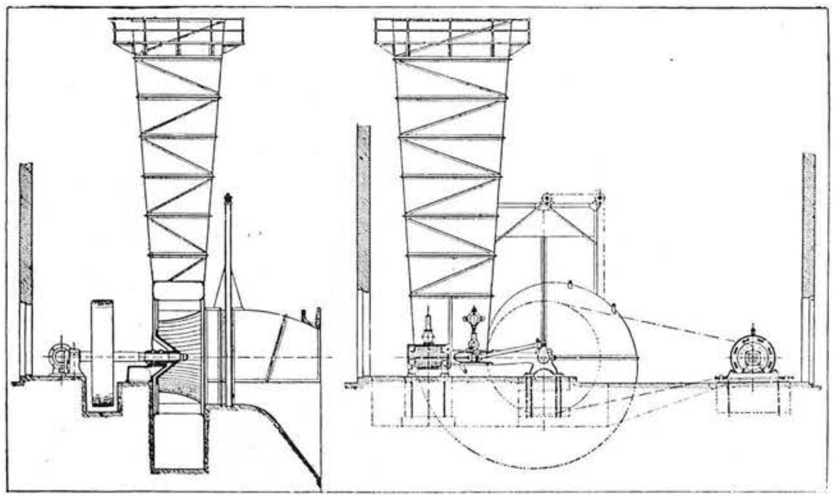

FIG. 62.—MONNET-&-MOYNE FAN, 4 METRES (13 FEET $1\frac{1}{2}$ INCHES) IN DIAMETER, AT THE LIÉVIN COLLIERIES, PAS DE CALAIS.

Particulars of Evasées of Colliery Fans.

Fan	Hade at:- Wall nearest fan. Degrees.	Wall farthest from fan. Degrees.	Side-walls. Degrees	Area of throat in square feet	Area of mouth in square feet.	Mouth area ÷ throat area.	Length in feet.	Remarks.
[G]uibal	Vertical	8	Vertical	27.0	45.60	1.70	18.0	Intended for 50,000 cub. feet per minute at 1.5-ins. of water-gauge. Installed 1865 from Guibal's designs.
[Do]uble-[in]let [Ca]pell, [ol]d type [a]bout [18]90).	12	12	17	108.0	168.00	1.55	6.0	
[Do]uble-[in]let [Ca]pell [r]ecent)	Vertical	15	6½	66.5	115.00	1.73	10.5	Intended for 300,000 cub. ft. at 4-ins. of water-gauge (see Trans. Inst. M.E., 1921-1922, vol. LXIII. page 107).
[Wa]ffrey	Vertical	12	8	-	-	2 (approx)	-	
[Si]ngle-[in]let [S]ith	Vertical	15	3½	17.5	43.75	2.50	13.5	Intended for 100,000 dub. feet at 7-inches of water-gauge (see Fig. 59.
[Do]uble-[in]let [Si]rocco	Vertical	12	Vertical	27.0	45.60	1.70	18.0	Intended for 400,000 cub. feet at 8-ins.of water-gauge (see Fig. 60
[Sc]hiele	Vertical	10	17	-	-	1.4 (approx)	6 (approx)	
[R]ateau	Vertical	7½	3¾	11.6	40.90	3.50	28.0	Maximum efficiency at 50,757 cubic feet and 3.3-ins. of water-gauge. Base of chimney curves through a quadrant.

Particulars of Evasees of Colliery Fans. (Continued).

Fan.	Hade at:-			Area of throat in square feet.	Area of mouth in square feet	Mouth area / throat area.	Length in feet.	Remarks.
	Wall nearest fan. Degrees.	Wall farthest from fan. Degrees.	Side-walls. Degrees.					
Monnet-& Moyne	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$	29.0	86.50	3.00	25.5	See Fig. 62.
Ditto.	4	4	4	18.2	78.50	4.30	32.0	
alland	4	4	4	-	-	3.00	-	See Fig. 61

directed to their évasées.

In 1880, Boulker and Watson introduced a somewhat ~~similar~~ novel form in évasée design.[1] By employing évasées at each of three equi-distant positions about the fan casing, they endeavoured to reduce the internal fan losses, by thus allowing the fan to discharge air at three points about its periphery. The évasées partly encircled the casing, so that the direction of the discharged air was not changed too abruptly. The angle of divergence was 7°. This fan, however, did not appear to meet with much success, although its inventors produced strong experimental evidence of its advantages.

Figure 63 shows a section of the annular discharge outlet of the Waddle fan installed at Wellesley Colliery; this fan is 24-feet in overall diameter, and is designed to give 210,000 cubic-feet of air per minute at a pressure of 26 pounds per square-foot. Air is discharged all round the periphery of the fan in this arrangement. The convergent part of the outlet is of recent design, and it is claimed that a considerable increase in efficiency has resulted from its use. Nevertheless, we have very good reason to believe that an enormous dissipation of kinetic energy occurs at the outlet of this type of fan.

In /

1. "An Account of a New Ventilating Fan", by T. J. Boulker, Trans. N.E. Inst. 1881-1882, Vol. XXXI, page 93.

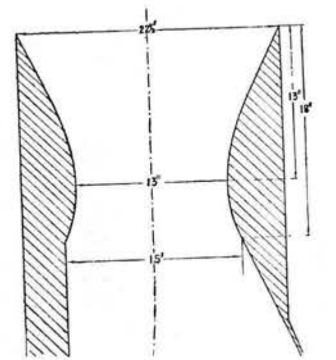

Fig. 63.—Section of Discharge Ring of 21-foot Waddle Fan.

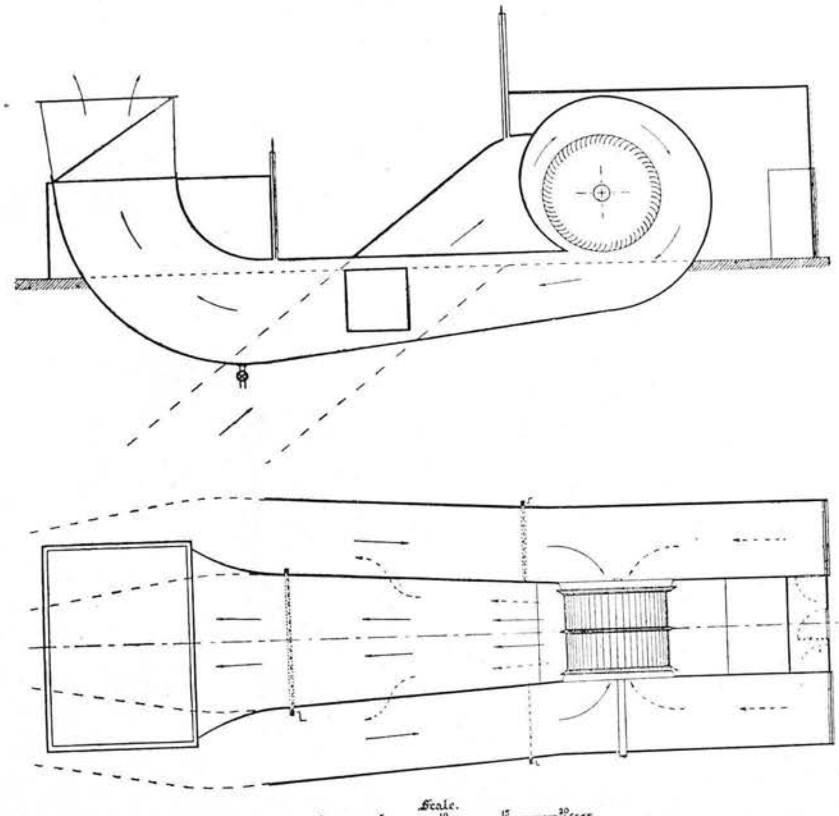

Fig. 64.—A Proposal to obtain a 4-to-1 Expansion without involving a large Vertical Chimney. The full Arrows indicate the Normal and the dotted Arrows the Reversed Direction of Flow.

In British designs, the area at the "throat" is generally too large to permit a four-to-one expansion such as we found to be the economical limit in length of évasée. For example, for the chimney of the Wellesley Sirocco, to have an apical angle of 7° and an expansion such that the mouth area was four times that of the existing throat area, the chimney would require to be 42-feet in height. The cost of such an erection would in all probability outweigh any improvement in efficiency which may accrue.

The Galland and Monnet & Moyne installations, as already pointed out, approach very nearly the design which we have experimentally determined to yield the maximum efficiency. This fact is a sufficient guarantee that the erection of évasées of efficient design is practicable. Figure 64 illustrates a method whereby the full four-to-one expansion at 7° could be obtained, even when the throat area is great, without the erection of a large upright chimney, as is the common practice. Only a small portion of the évasée need be vertical, the greater part being a horizontal passage, the roof of which coincides with ground-level; the walls and roof of such passage may be built of concrete. Since the velocity of the air will be greatly diminished, no serious objection can be raised against the right-angle bend. The reversal of the air-current could be readily effected by the door arrangements shown. The provision of a cock at the lowest point of the évasée would allow any accumulation of water to drain into the fan drift. Modifications of this proposal could easily be made to suit existing conditions. For instance, it might be better in some cases to make the horizontal part of the évasée in the opposite direction to that shown in Figure 63, and connect such to the fan drift with a by-pass for reversal. For single-inlet fans, where the fan-drift leads directly into the fan- inlet, the horizontal expansion could be made in a direction normal to the fan-drift.

(12) Conclusions, (Section (c)).

(a) Pressure energy in air can be transformed into kinetic energy by any uniformly converging duct with an efficiency of almost 100 per cent. In such ducts, stream-line flow, or a condition closely approaching it, can be maintained up to a high velocity.

(b) Kinetic energy in air can be efficiently trans:formed into pressure energy by diverging ducts only over a limited range of apical angles. The maximum efficiency attainable in this conversion does not approach that so easily obtained in the reverse operation.

(c) Irregularity of flow in expanding ducts is best reduced, and re-entry avoided by decreasing the angle of divergence. Energy loss due to skin friction is almost negligible compared to the loss in energy arising from re-entry and turbulent flow.

(d) Until a better distribution of air leaving a parallel-sided fan is effected, it is futile to expect uniform flow, or a condition even approaching it, at the throat of an évasée directly connected to the casing-outlet.

(e) The limiting angles for efficient divergence in évasées appear to be:-

(i) when all four sides expand uniformly, 5° and 9°; i.e., the hade of each side should not be less than $2\frac{1}{2}^\circ$ or greater than $4\frac{1}{2}^\circ$.

(ii) when two sides only expand uniformly and the other two are parallel, or when three sides are vertical and the other is inclined, 8° to 14°; i.e., in the first case, the hade of each of the two sloping walls should not be less than 4° or more than 7°, and where one wall only is inclined the limits of hade are 8° and 14°.

(f) The height or length of an évasée diverging according to (e) should be such that the mouth area is to the throat area as four is to one. The chimney need not be a vertical erection; most of the divergence could be made in a horizontal direction as suggested in Figure 64.

(g) The practical ideal in fan évasée design would appear to be one in which the wall adjacent to the fan was straight, the wall most remote from it was inclined at 7^{o}, the two side-walls each haded at $3\frac{1}{2}^{o}$, and the height or length was such that the four-to-one expansion was obtained.

(h) An évasée hading equally on all four sides yields, at its best, a higher efficiency than one expanding on two sides only; when the apical angle exceeds 11^{o}, the two-sided expansion may be better than the four-sided one.

(i) The maximum efficiency attainable with an évasée designed in accordance with a 7^{o} divergence is, under the most favourable conditions, 80 per cent.

(j) The greatest loss of energy in évasées occurs at their commencement. The condition of the inflowing air and the shape of the évasée near the throat, are of primary importance.

(k) The perfect évasée would be one of circular cross-section and designed in accordance with the <u>vena contracta</u>; the converging portion would produce stream-line flow prior to the air entering the divergent part. The mean dynamic gauge-reading in a perfect évasée would be uniform throughout its length.

(l) A badly designed évasée, or a parallel-sided discharge duct, is better than no évasée at all; the non-expanding outlet is conducive to increased regularity in the velocity of discharge and hence minimises the dissipation of energy. The efficacy of a short length of parallel-sided duct interposed between the casing outlet and the divergent discharge requires further investigation.

(m) An évasée of the best design can deal efficiently with an entrant air-velocity as high as 230-feet per second. (This was the velocity of the air entering /

entering the évasée which gave the highest efficiency).

(n) In fans provided with a large "volute", and hence a large évasée throat area, a considerable amount of kinetic energy is lost in the "volute" due to friction between air-streams differing in both direction and velocity. Where no "diffusion" arrangement exists, it would seem preferable to have the casing such that little or no decrease in the velocity of air emerging from the fan occurred until it entered the évasée, i.e., to leave the whole of the conversion of kinetic into pressure energy to the évasée.

(p) Once erected, little if any of the saving effected by an évasée is ever required for maintenance costs. With a large installation, the difference between an efficient évasée and an ill-designed one, may ~~never~~ mean the manager's salary.

PART IV - SECTION D.

FAN CASINGS.

(1) Introductory.

Our study of fan casing design is incomplete. Nevertheless, it is considered that the experimental investigation so far conducted may in some slight measure effect improvement in the design of this important component of the ventilator.

The evolution of the fan casing was briefly traced in Part II (See page 12). In previously recorded experimental work we had occasion to comment upon the influence of the casing in connection with the elimination of centripetal re-entry. Again, in the preceding section, the efficacy of large volute casings, and incidentally large évasée "throat" areas, was doubted.

(2) Modern Fan Casings.

Guibal's concentric form of casing while still retained in fans of that name has otherwise been superseded by the more rational form, the volute or spiral. Figure 65 illustrates the outline of the casing belonging to the laboratory 18-inch Sirocco fan; it is a replica of the form of casing used in the larger Sirocco installations. Apart from its distinctive shape at the commencement, where it is straight for some distance, this form of casing is representative of most of the existing British designs.

In general, the casing is made voluminous enough so that besides functioning as the collecting and guiding medium of the air discharged from the fan wheel to the évasée, it also effects a partial conversion of kinetic energy into pressure energy, the remainder of the transformation being left to the évasée. When a slightly divergent diffuser is interposed between the fan-runner and the casing, an efficient partial recovery of kinetic energy may be expected, but in other cases, it seems highly /

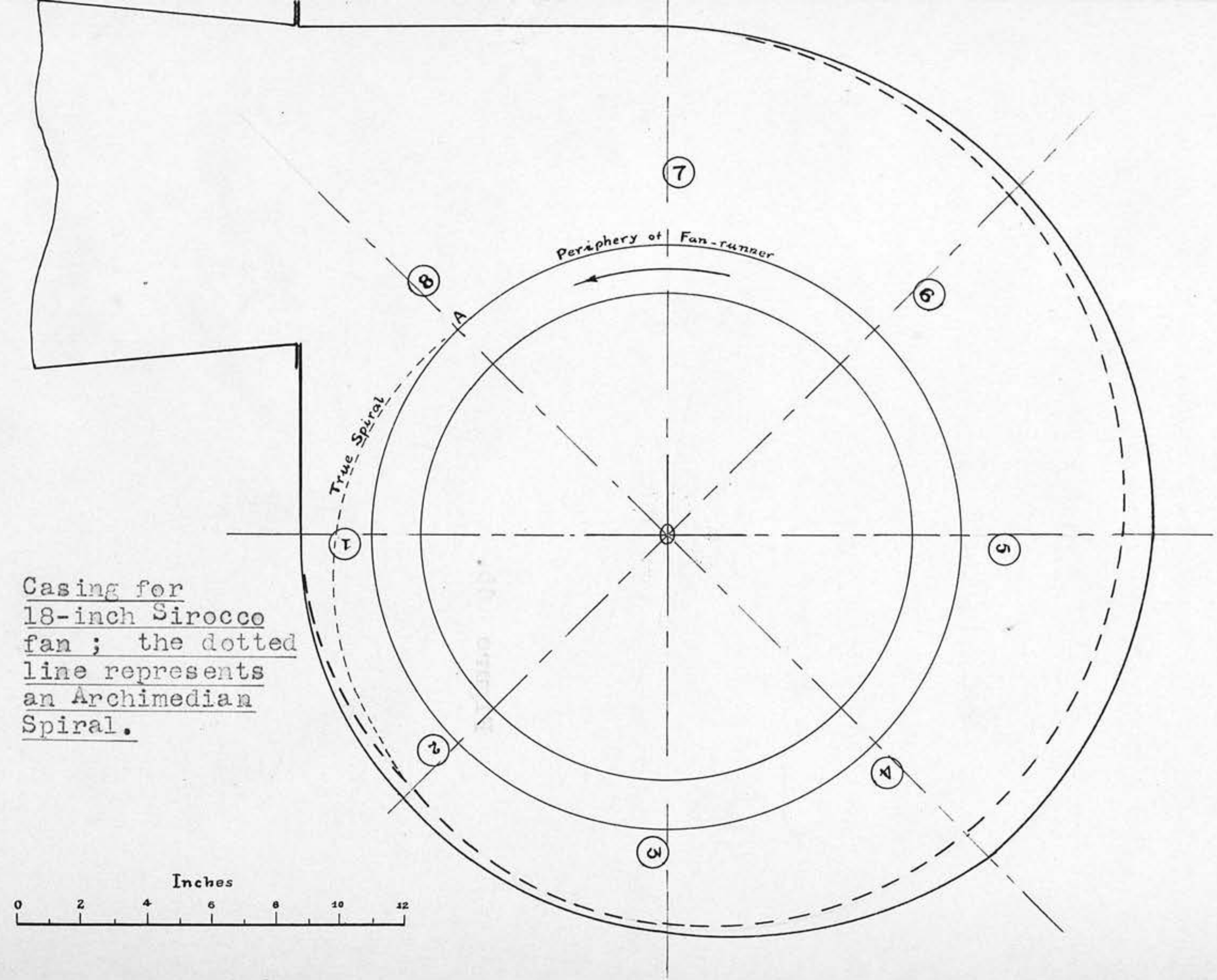

Figure 65. Casing for 18-inch Sirocco fan ; the dotted line represents an Archimedian Spiral.

— Figure 65. —

highly probable that any energy transmutation effected is obtained only at great loss, largely due to collisions between air-streams of varying direction and velocity. British designs are almost entirely parallel-sided, and of a width considerably greater than that of the fan wheel. The width of the Sirocco casing, for example, is $1\frac{2}{3}$ times that of the fan-runner. The parallelism of the sides, considered alone, does not appear to be a bad feature, but the sudden increase of width cannot be conducive to the efficient conversion of velocity energy.

The volute part of the casing is manufactured in a series of arcs, the maker's endeavour being to complete the volute with as few radii as possible. While such a course may facilitate construction, it must frequently lead to greater or less departures from the path of a true volute or spiral. In Figure 65, the true Archimedian spiral to give the same depth of section at the outlet as exists, is shown by the dotted outline. The difference in the two curves is apparently slight; nevertheless, in tests shortly to be described it is shown that such difference is sufficient to produce an adverse influence on the efficiency of the plant. The equation for the spiral curve in Figure 65 is:-

$$x = k\alpha + a \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (25)$$

where x = the radius from the fan-centre to the casing at any point.

α = the angle between OA and x, in radians.

a = the radius of the fan runner.

k = a constant (1.275 in our case).

In the Rateau fan, the formula giving the section (S) of the volute as functions of the arc of the diffuser spiral is,

$$S = 0.5x\,(1 + 0.8x)\,O \ldots\ldots\ldots\ldots\ldots\ldots\ldots (26)$$

where S = the cross-section of the volute at any point.

x = is the ratio of the length of the arc of the diffuser spiral from the origin to the point at which the section is to be calculated to the total length of this spiral.

O = the area of the fan inlet.

There are a few special features associated with the casings of certain fans which are perhaps worthy of mention. Reference to the Guibal adjustable shutter was made on page 13. The area of the peripheral discharge from the Guibal is regulated by means of this shutter, the lower edge of which is shaped like a swallow's tail to prevent the humming noise which was characteristic of the original design. In the Walker fan, which is a modernised Guibal, a similar shutter arrangement is employed for varying the discharge outlet. In Figure 38, (opposite page 91) the Mortier fan is illustrated; the section of the by-pass B can be varied, as indicated by the dotted lines, to suit various conditions.

(3) Previous Experimental Work related to Fan Casings.

The field for review is indeed meagre. Guibal himself records experiments conducted by Mon. Gille and Franeau at Frameries (Belgium) on one of his ventilators when running (1) open, (2) with cover, (3) with cover and évasée, and (4) with cover, évasée and shutter.[1] The results obtained are summarised below:-

	Minimum Useful Effect.	Maximum Useful Effect.	Mean Useful Effect
1. Without casing	0.16	0.22	0.19
2. With casing	0.09	0.31	0.20
3. With casing and évasée	0.26	0.57	0.415
4. With casing, évasée and shutter.	0.38	0.61	0.495

The general conclusions arrived at regarding the efficacy of the casing were that, uncovered, the fan was very inefficient; with a cover "badly arranged", it could be more useless than useful; and below certain speeds the cover lowered the efficiency.

Figure 66 is a diagram which was drawn up by Belgian Government officials about 1870, showing the variation in depression /

1. "On Some Experiments with the Covered Ventilator of M. Guibal", Communicated by A.L.Steavenson. T. N.E. Inst. of

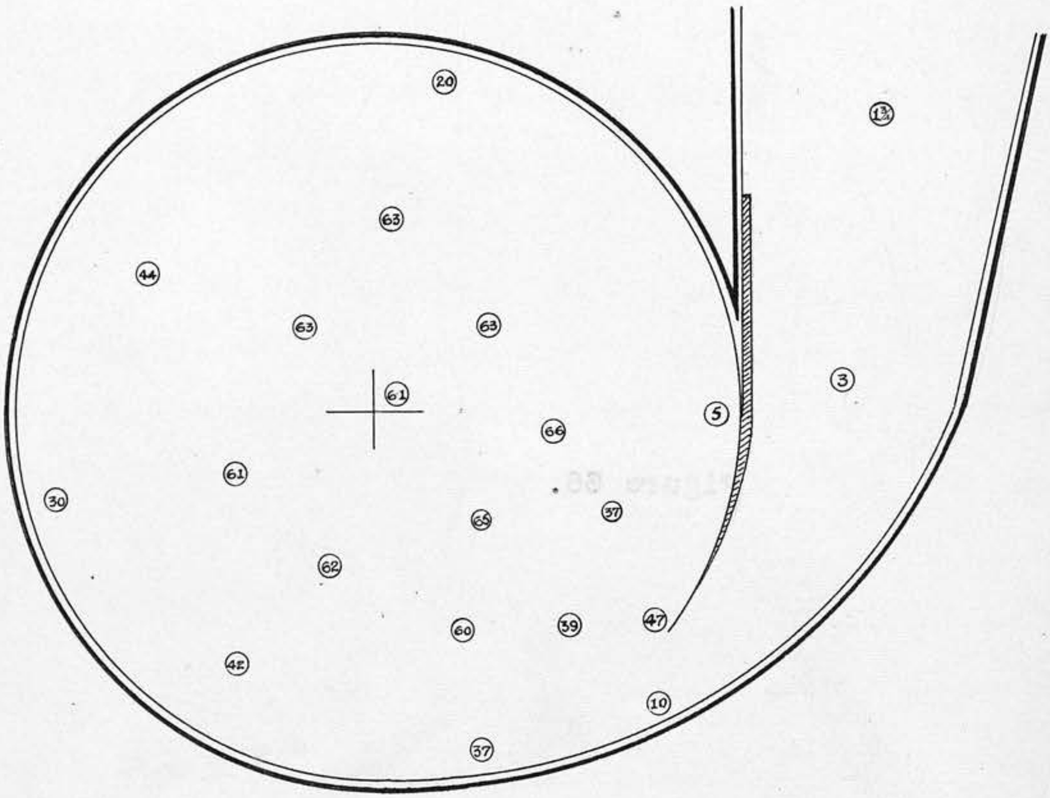

Figure 66. Diagram showing Depressions (in millimetres of water) inside the Casing of a Guibal Ventilator at Work.

depression inside the casing of a Guibal Ventilator.[1]

The experiments of Donkin and Heenan and Gilbert have been already reviewed.

(4) <u>Objects of Experimental Investigation.</u>

The experimental work which we have so far conducted concerns the casing of the 18-inch Sirocco fan shown in Figure 65. The objects of the investigation were to determine:-

(i) the extent of the variability in velocity of flow round the fan casing;

(ii) what improvement, if any, would be effected by the substitution of a casing truly spiral in form for the existing one;

(iii) the effect of varying the clearance between the fan runner and the commencement of the casing or "beak" of the fan.

(5) <u>Apparatus Used in the Investigation.</u>

For the detailed measurement of velocity round the casing an attempt has been made to devise an electrical hot-wire anemometer which would give direct measurements of both velocity and direction, and at the same time would not greatly disturb the normal motion of the air-streams. In hot-wire anemometry, there are two principle methods used; in one, the temperature of the wire exposed to the flowing air (and therefore its resistance) is kept constant by varying the current, the value of which affords a measure of the rate of air-flow past the hot-wire; in the other method, a constant current is passed through the wire and its resistance determined by means of the galvanometer, the deflections of which can be calibrated to represent rate of air-flow. The latter method was the one used in our experiments.

The design of the instrument so far developed is principally due to Professor Briggs. Figure 67 diagrammatically illustrates the arrangement of the first device /

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1. From Boulker's paper, op. cit., Vol. XXXII, p. 26. A. L. Steavenson in discussion.

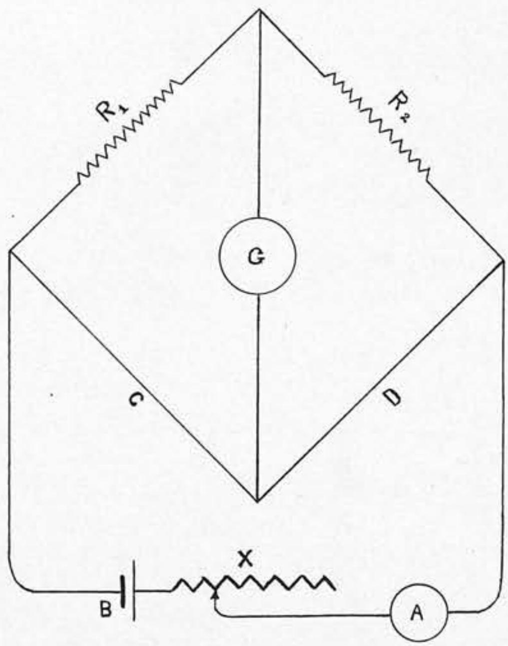
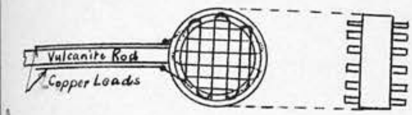


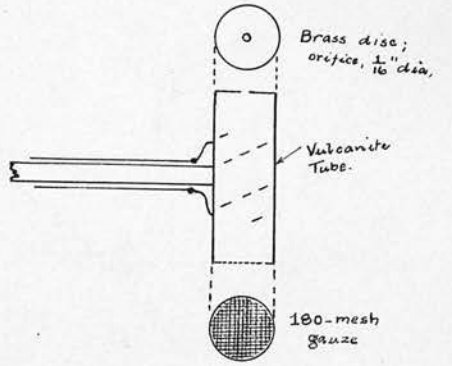
Figure 67. : Diagram showing the first arrangement
the electrical hot-wire velocity-meter

References:-

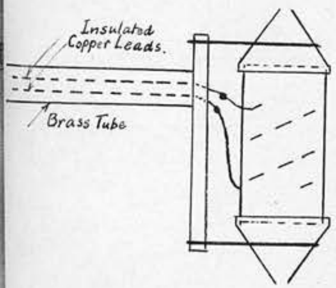
R_1 and R_2;	Standard resistances, each 1000 Ohm
C;	length of pure nickel wire (S.W.G. temp. co 0.006 p
D;	same as C; used in anemometer-head.
G;	galvanometer.
B;	accumulator supplying a steady curr
X;	rheostat.
A;	ammeter.



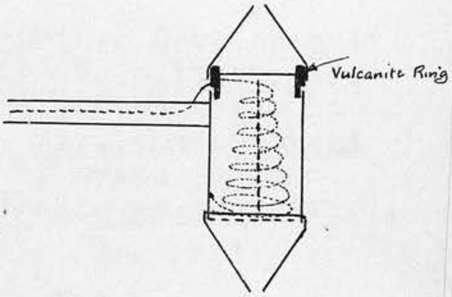
(a) Nickel wire arranged in form of Grid on Vulcanite Tube.



(b) Nickel wire arranged as a star (end view) by threading it through 12 different diameters of the vulcanite tube; brass disc with small orifice at one end of tube and 180-mesh gauze at other.



(c) Nickel wire arranged as in (b); thread-holes in vulcanite filled up with gypsum;
Brass cones with small orifices fitted to ends of tube.



(d) Nickel wire arranged in series of decreas:ing spirals on sheet of mica fixed axially inside vulcanite tube. Conical end-pieces can be changed.

Figure 68.: Forms of Anemometer-heads used in Experiments.

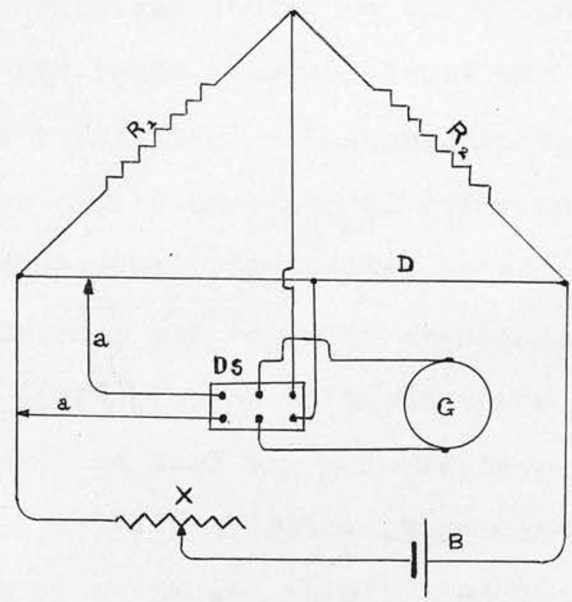


Figure 69.: Diagram illustrating a further Development of the Electrical Hot-Wire Velocity Meter. (In this arrangement, the galvanometer is used for a dual purpose, viz., measurement of change in resistance of D-and hence velocity of air-flow-and also as an ammeter).

References:-

R_1, R_2, D, G, B and X as in Figure 67.

C; manganin wire equivalent to resistance of D in still air.

~~S; switch.~~

D.S; double knife-blade switch.

a,a; connections when G is used as an ammeter.

device tried, full references being given below the Figure. The fine nickel wire forming the arm D, (and the anemometer) was arranged in the form of a grid on a short length ($\frac{1}{4}$ inch) of vulcanite tube, $\frac{3}{8}$ - inch diameter (see Figure 68 (a)). In this form the instrument was very sensitive indeed to the slightest movement of air but its range of snesitivity was limited to comparatively low velocities. The velocities which we desired to measure easily exceeded 60-miles per hour, and since the Macgregor-Morris instrument referred to on page 46 is useless when the velocity approaches 5-miles per hour, the difficulty to be encountered in the design of an instrument to suit our purpose is perhaps sufficiently apparent. Various forms of anemometer-heads have been tried some of which are illustrated in Figure 68. In the latest design (d), the nickel wire is not allowed to touch the vulcanite at any point, being wound in decreasing spirals on a strip of mica fixed axially inside the vulcanite tube; by varying the orifices in the conical heads varying ranges of velocity can be measured. The greatest trouble experienced has been the "zero" check. This was found to be most erratic with the earlier forms, and while not yet completely solved, the difficulty is being gradually overcome. Figure 69 shows a further development of the instrument; while still unreliable, unless calibrated before and after a series of readings in the fan casing, it is nevertheless capable of measuring velocities as high as 90 miles per hour.

This device is being further developed, a new arrangement of the Bridge and the improved design of head (d) are to be tried, and it is hoped that ultimately an anemometer capable of measuring a wide range of velocities with a fair degree of accuracy will be evolved.

In /

In consequence of its present state of unreliabil::ity, the electrical hot-wire device was not used throughout the detailed exploration made inside the fan casing; it was occasionally employed, however, as a check on the "Pitot" tube measurements. The form of Pitot tube used throughout this part of the investigation is illustrated in Figure 70; it resembles the B.T.H. Steam Meter, and consist of two brass tubes each $\frac{1}{16}$ - inch diameter, and 11-inches long between the right-angle bends. As will be observed, it is not a true form of Pitot tube, since a suction head and not the true static head is obtained at the trailing orifice. This instrument was carefully calibrated against the zero-setting anemometer, the latter itself being first calibrated in the manner already specified. The correction factor for the "Pitot" tube was found to be 0.88, or

$$\text{Velocity (in feet per minute)} = 966\sqrt{\frac{h}{w}}$$

where h is the velocity head (in inches of water-column) recorded by the "Pitot", and w the weight of a cubic foot of air under the prevailing conditions.

The inclined manometer previously described was used in conjunction with this improvised Pitot tube. In the efficiency tests, an ordinary dynamic tube was employed in the manner followed in the earlier work. Frequent measurements of atmospheric conditions were also taken to allow w to be determined. Ordinary air velocities in the fan-drift were measured with the zero-setting anemometer.

(6) Detailed Measurement of Velocity of Air inside Casing of 18-inch Sirocco Fan.

Throughout this exploration, the fan speed was maintained at 750 revolutions per minute, i.e., 3531 feet per minute. The mean volume of air discharged by the fan was 1647 cubic feet per minute against a mean pressure of 6.34 pounds per square foot. At the sections of the fan casing numbered in Figure 65, holes were bored at various depths large enough to admit the head of the hot-

wire /

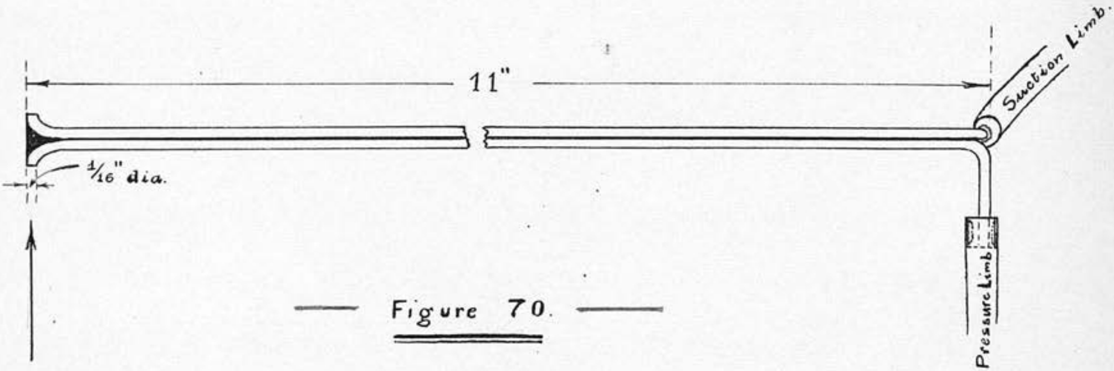


Figure 70.

Form of Pitot-tube used in Fan Casing Experiments

Figure 71. Developed Diagram (not to scale) showing Relative Air Velocities (in feet per minute) round Periphery of 18-inch Sirocco Fan when running at 3,531 feet per minute and passing 1,647 cubic feet of Air in the same time.

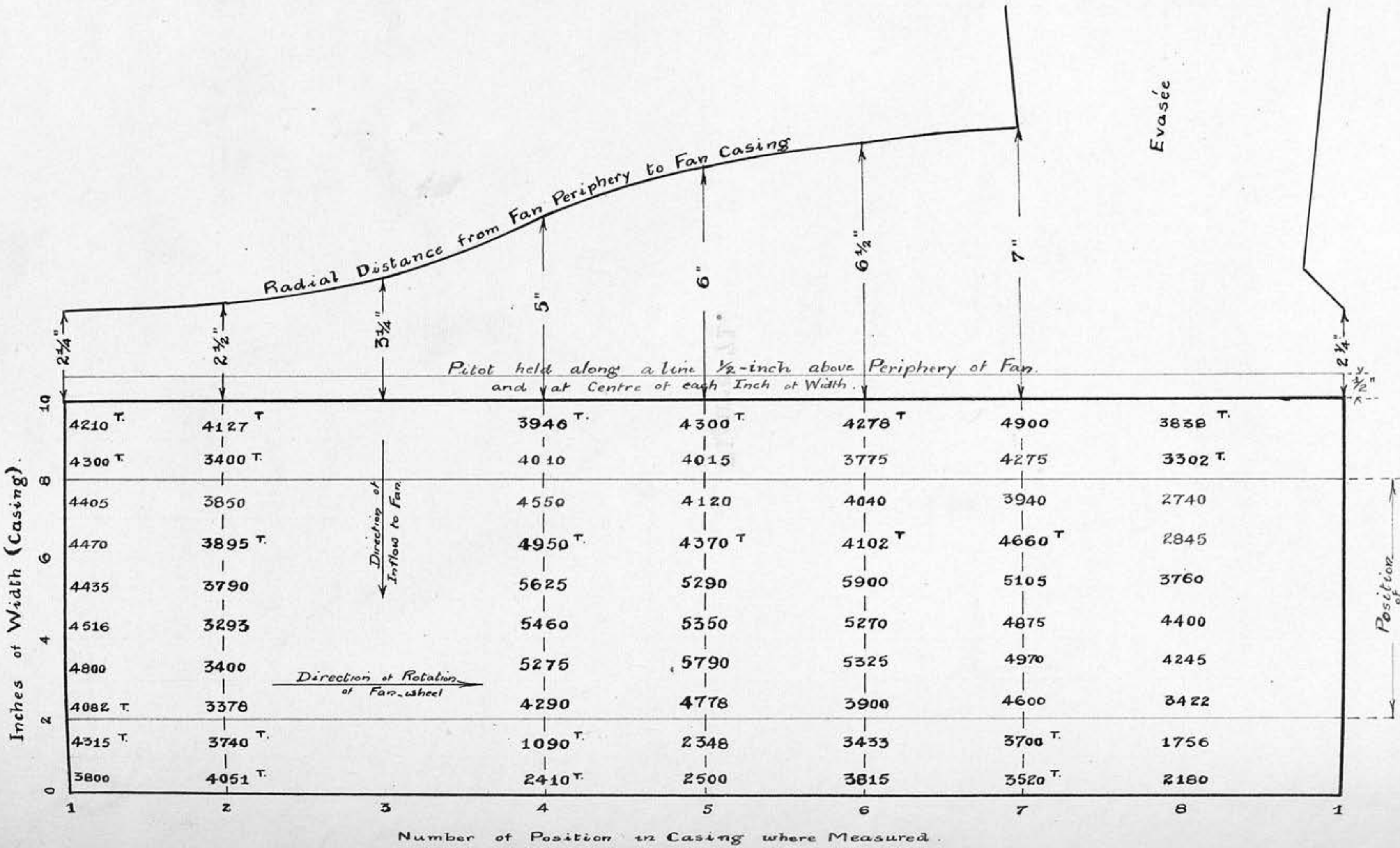
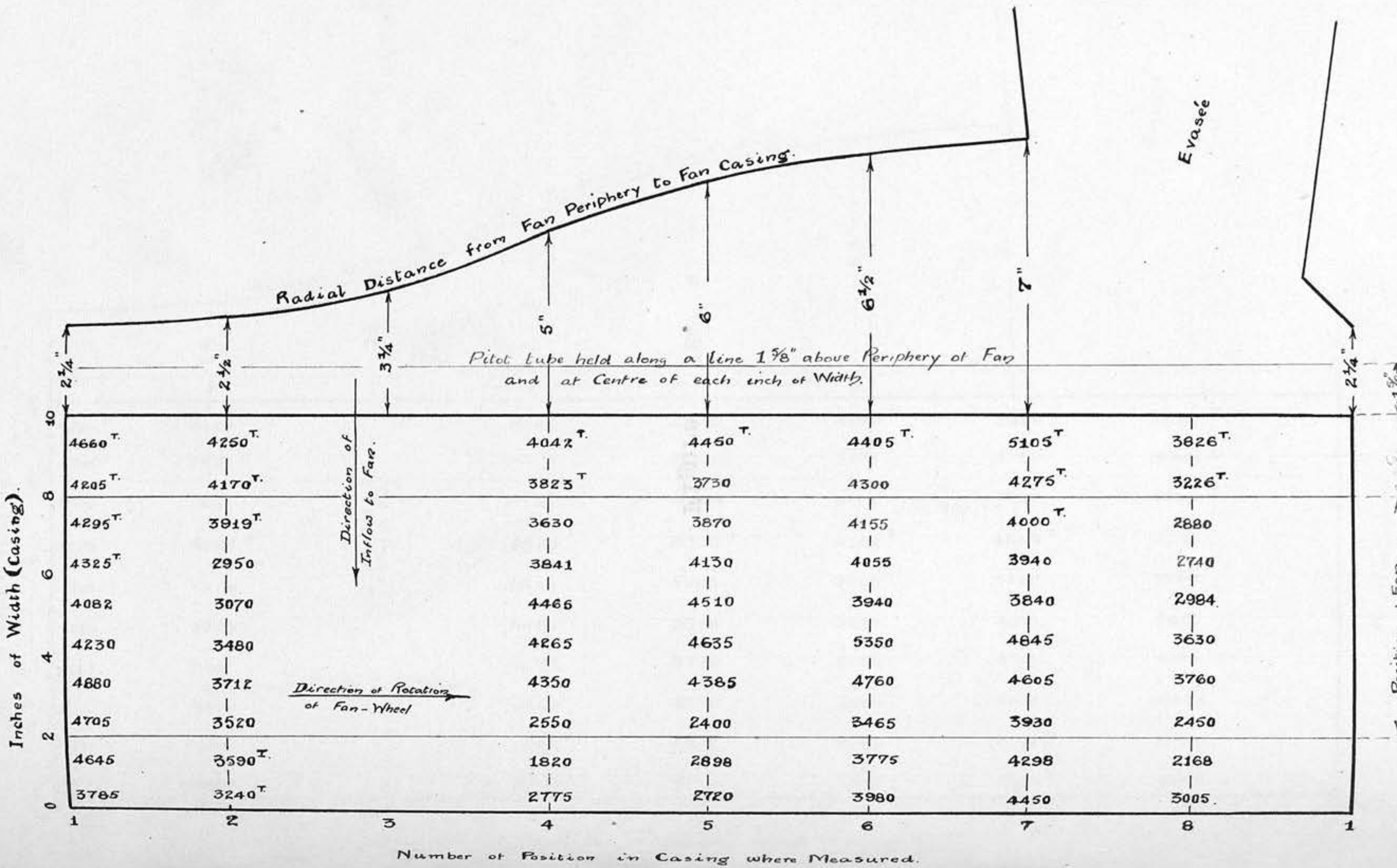
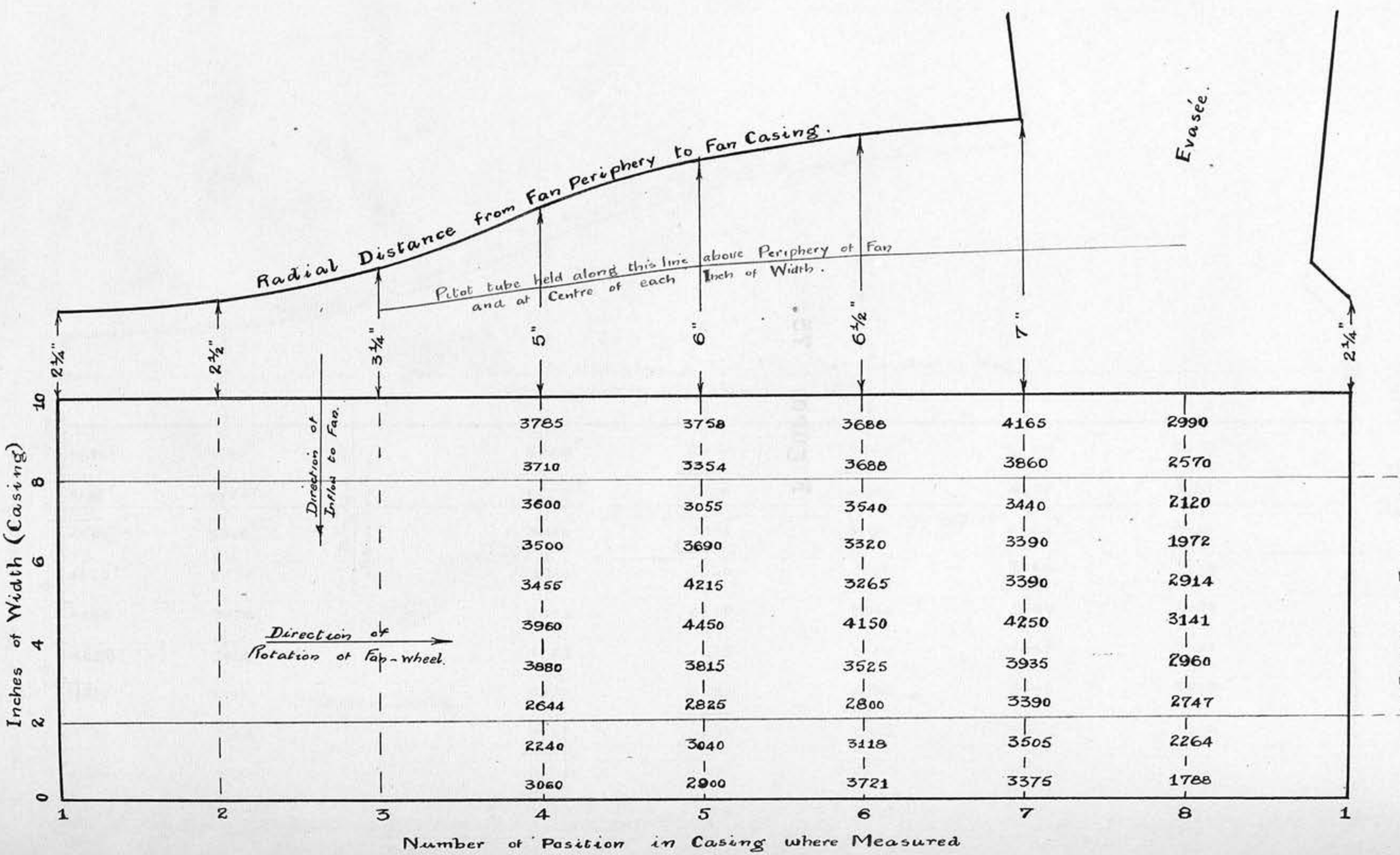


Figure 72.



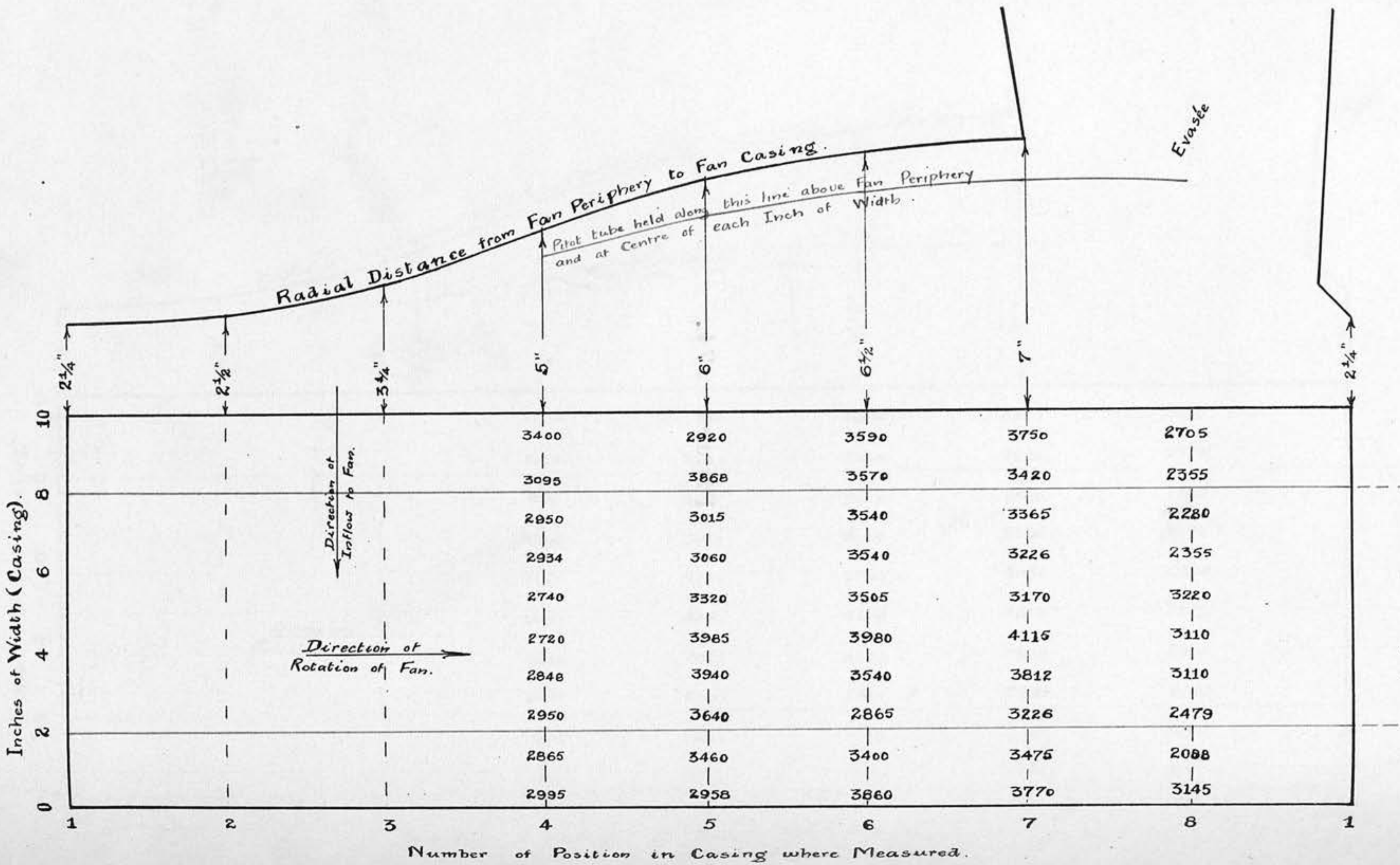
Radial Distance from Fan Periphery to Fan Casing.
2¼"
2½"
3¾"
5"
6"
6½"
7"
2¼"
Evaseé
Pitot tube held along a line 1⅝" above Periphery of Fan and at Centre of each inch of Width.
1⅝"
Position of Fan
Direction of Inflow to Fan.
Direction of Rotation of Fan-Wheel
Inches of Width (Casing).
10
8
6
4
2
0
1
2
3
4
5
6
7
8
1
Number of Position in Casing where Measured.
4660 T. 4205 T. 4295 T. 4325 T. 4082 4230 4880 4705 4645 3785
4250 T. 4170 T. 3919 T. 2950 3070 3480 3712 3520 3590 T. 3240 T.
4042 T. 3823 T. 3630 3841 4466 4265 4350 2550 1820 2775
4450 T. 3730 3870 4130 4510 4635 4385 2400 2898 2720
4405 T. 4300 4155 4055 3940 5350 4760 3465 3775 3980
5105 T. 4275 T. 4000 T. 3940 3840 4845 4605 3930 4298 4450
3826 T. 3226 T. 2880 2740 2984 3630 3760 2450 2168 3005

Figure 73.



Radial Distance from Fan Periphery to Fan Casing.
Pitot tube held along this line above Periphery of Fan
and at Centre of each Inch of Width.
Evasée.
2¼"
2½"
3¼"
5"
6"
6½"
7"
2¼"
Direction of Inflow to Fan.
Direction of Rotation of Fan-wheel.
3785
3710
3600
3500
3456
3960
3880
2644
2240
3060
3758
3354
3055
3690
4215
4450
3815
2825
3040
2900
3688
3688
3540
3320
3265
4150
3525
2800
3118
3721
4165
3860
3440
3390
3390
4250
3935
3390
3505
3375
2990
2570
2120
1972
2914
3141
2960
2747
2264
1788
Inches of Width (Casing)
0
2
4
6
8
10
1
2
3
4
5
6
7
8
1
Number of Position in Casing where Measured

Figures 74.



4	5	6	7	8
3400	2920	3590	3750	2705
3095	3868	3576	3420	2355
2950	3015	3540	3365	2280
2934	3060	3540	3226	2355
2740	3320	3505	3170	3220
2720	3985	3980	4116	3110
2848	3940	3540	3812	3110
2950	3640	2865	3226	2479
2865	3460	3400	3475	2088
2995	2958	3860	3770	3145

wire anemometer or the "Pitot" tube. At each position and depth of section, ten observations of velocity were taken, corresponding to each inch of width of the fan casing. At each position, the velocity recorded by either instrument was found to vary considerably with the least rotational movement. While no serious attempt was made to obtain direction together with the velocity measurements, the "Pitot" tube was slightly rotated in a vertical plane parallel to the main direction of flow, and the maximum reading taken for each position. No observations were made across the section of casing at position No. 3; the proximity of the fan-motor made access to this position difficult.

In the four developed diagrams - Figures 71 to 74 - the results are set out in such a way that they may be ~~more~~ readily visualised; the peripheral areas of fan and casing are relatively shown, together with the varying depth of the complete casing section. The direction of inflow to the fan is indicated on each diagram, as is also the direction of rotation. The figures in red ink represent velocity of air re-entering the fan at the particular positions. All velocities are given in feet per minute. The letter T beside a figure indicates that the conditions were very turbulent in that region.

The "Pitot" tube was held along a line of width $\frac{1}{2}$-inch above the periphery of the blades at each of the 7 positions in the first circuit of observations (Figure 71). In the second, (Figure 72), the height above the blades was increased to $1\frac{5}{8}$-inch for each position. The lines of measurement for the remaining circuits are indicated in their respective diagrams (Figures 73 and 74).

The results are also given in Appendix E in a different manner; there, they are set out as divisional velocities for each of the seven sections of the casing considered.

From /

From the developed diagrams (Figures 71 to 74), the variability in velocity over any cross-section is pronounced. In general, the greater velocities were found in the region corresponding to the width of the fan found most effective in the distribution tests. The greatest velocity recorded occurred near the centre of the fan at position No. 6 in the first circuit, i.e., ½-inch above the fan blades; this velocity was 5900 feet per minute, or 74.6 per cent. in excess of the circumferential speed of the fan. The resultant velocity, determined ~~from~~ in the manner indicated on pages 23 and 24, is 4480 feet per minute, or 26.9 per cent. greater than the fan's peripheral velocity.

At position No. 2 and circuit No. 1 (nearest the fan runner) the velocities recorded over the innermost three inches of blade width were actually less than the circumferential speed of the fan. This position coincides with that indicated in Figures 44 and 45 where the centripetal re-entry occurred. The quantity passing the casing section at position No. 2 was 113 cubic feet per minute less than that measured from the divisional velocities at position No. 1. It would thus appear that a severe throttling action was taking place at position No. 2.

The total quantity passing the section of the casing at position No. 7, calculated from the divisional velocity measurements was 1873 cubic feet per minute; the quantity actually discharged by the fan was 1647 cubic feet per minute, a difference of 226 cubic feet. At position No. 8 - nearly opposite the "beak" of the fan and a region of extreme turbulence - the quantity passing the section of the casing one inch high, measured from the ten "Pitot" tube readings taken along a line ½-inch above the fan's periphery at that position was 224 cubic feet per minute. From the direction of flow indicated /

indicated by the "Pitot" tube, and confirmed by a pith ball, the whole of this quantity was re-entering the casing. Deducting this figure from the total quantity passing position No. 7, the quantity discharged would then be 1649 cubic feet per minute.[1]

With few exceptions, the velocity of the air between the periphery of the fan and the surface of the casing, was considerably reduced and thus a large proportion of the kinetic energy in the air emergent from the wheel was converted into some other form of energy. This decrease in velocity over a cross-section of the fan casing is more readily seen in Appendix E. Calculated from the averages of all the velocities observed (1) nearest the fan periphery, and (2) nearest the casing, the kinetic energy in the air was (1) 161.4 foot-pounds per second and (2) 114.5 foot-pounds per second, or a decrease of 29 per cent. The equivalent gain in static pressure calculated from the same figures was 1.7 pounds per square foot.

(8) Comparison between the Existing Casing of 18-inch Sirocco Fan, and a Truly Spiral Casing.

A spiral casing, made from sheet tin and secured to wooden distance-pieces bolted to the existing casing, was fitted as shown in the dotted outline in Figure 65; to begin with, the same shape of "boat" was maintained, i.e., the spiral commenced from a straight line. Careful overall efficiency tests were carried out and the results compared with those obtained when the ordinary casing was in use. The averages of several tests are shown graphically in Figure 75, overall efficiencies //

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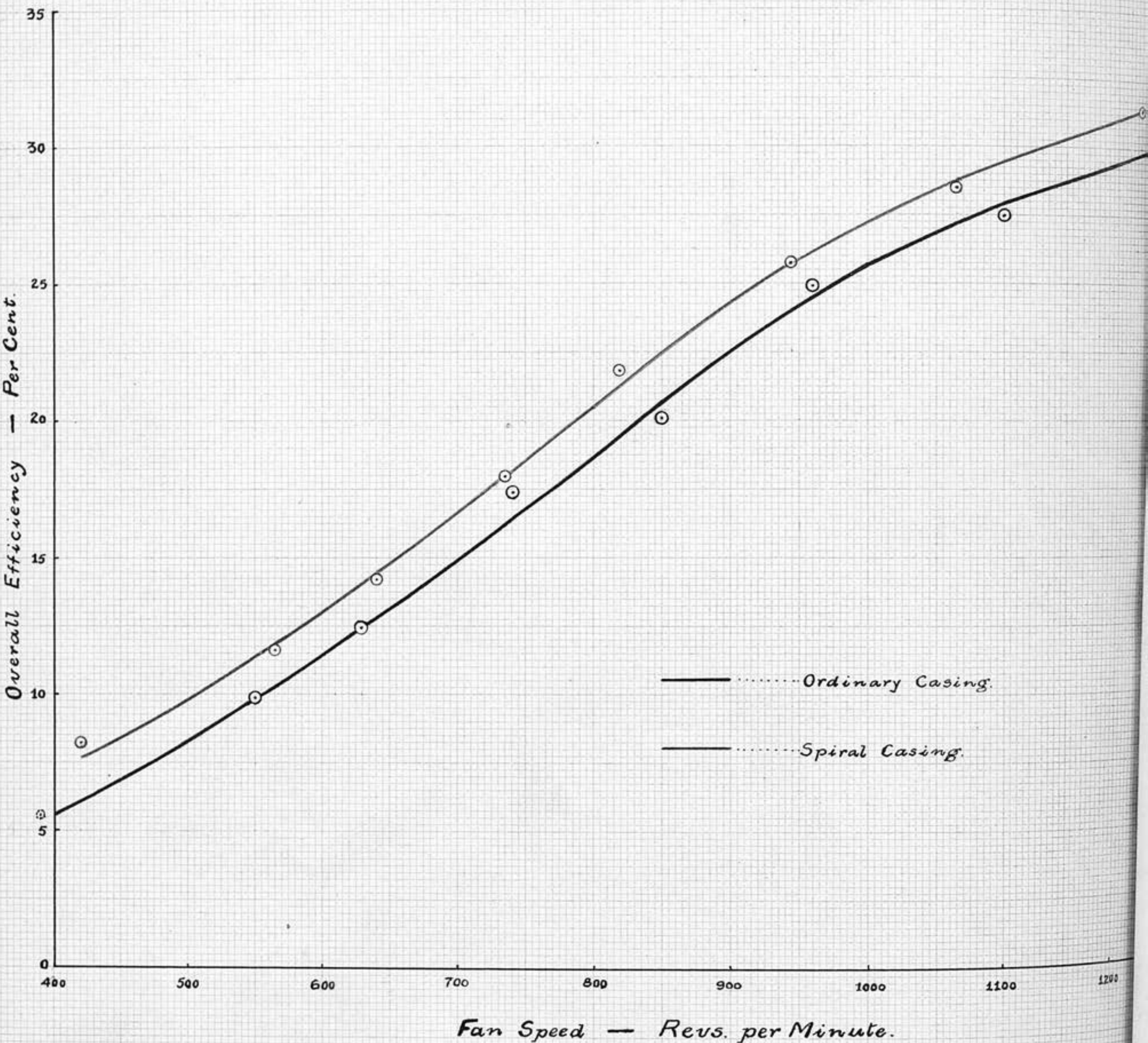
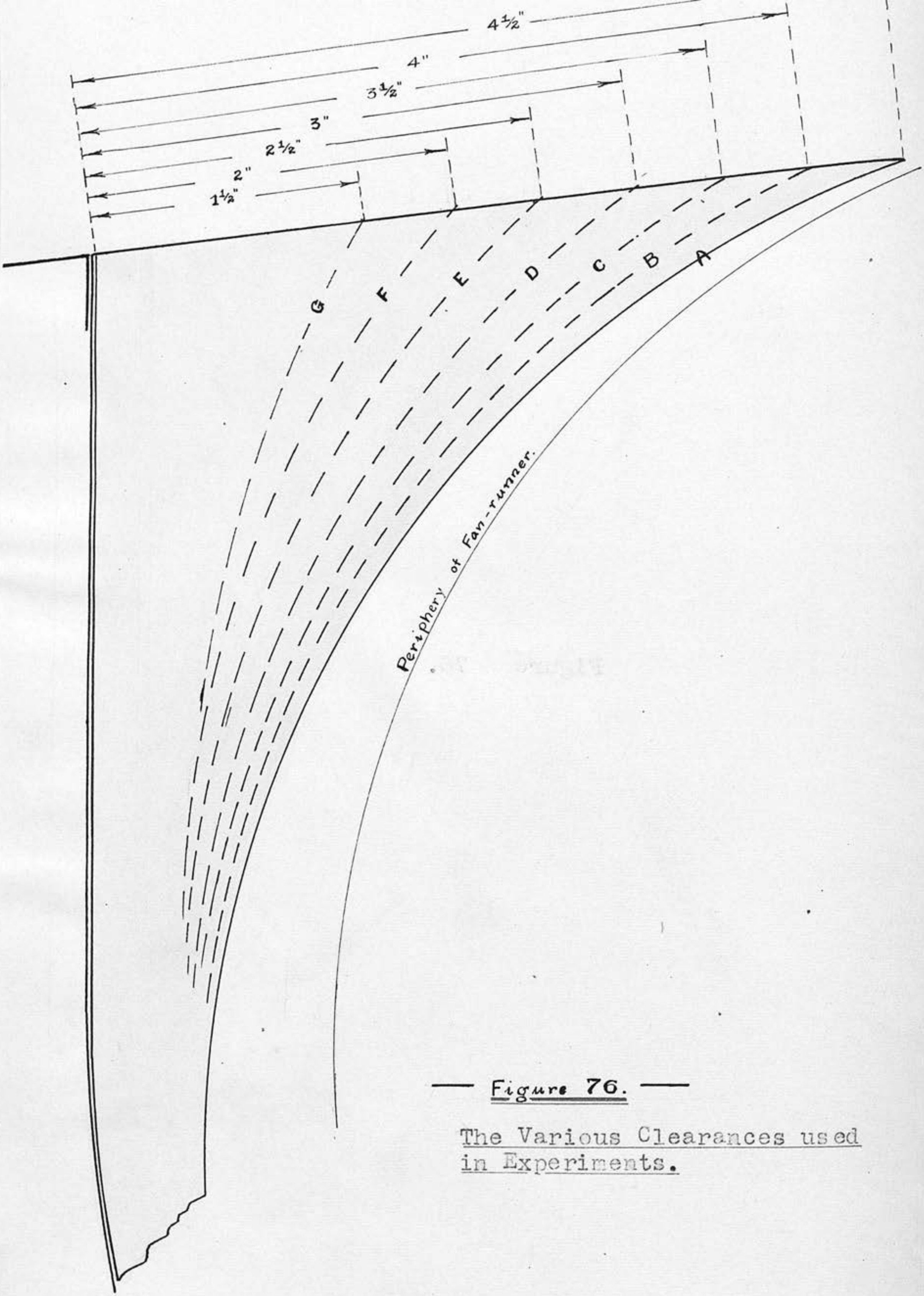
1. The closeness of agreement between the quantity measured by the anemometer and that derived from the divisional measurements is perhaps more of a coincidence than anything else; the main point at issue is the large percentage of air being re-circulated by the fan.

Figure 75. Efficiency Curves of 18-inch Sirocco Fan.

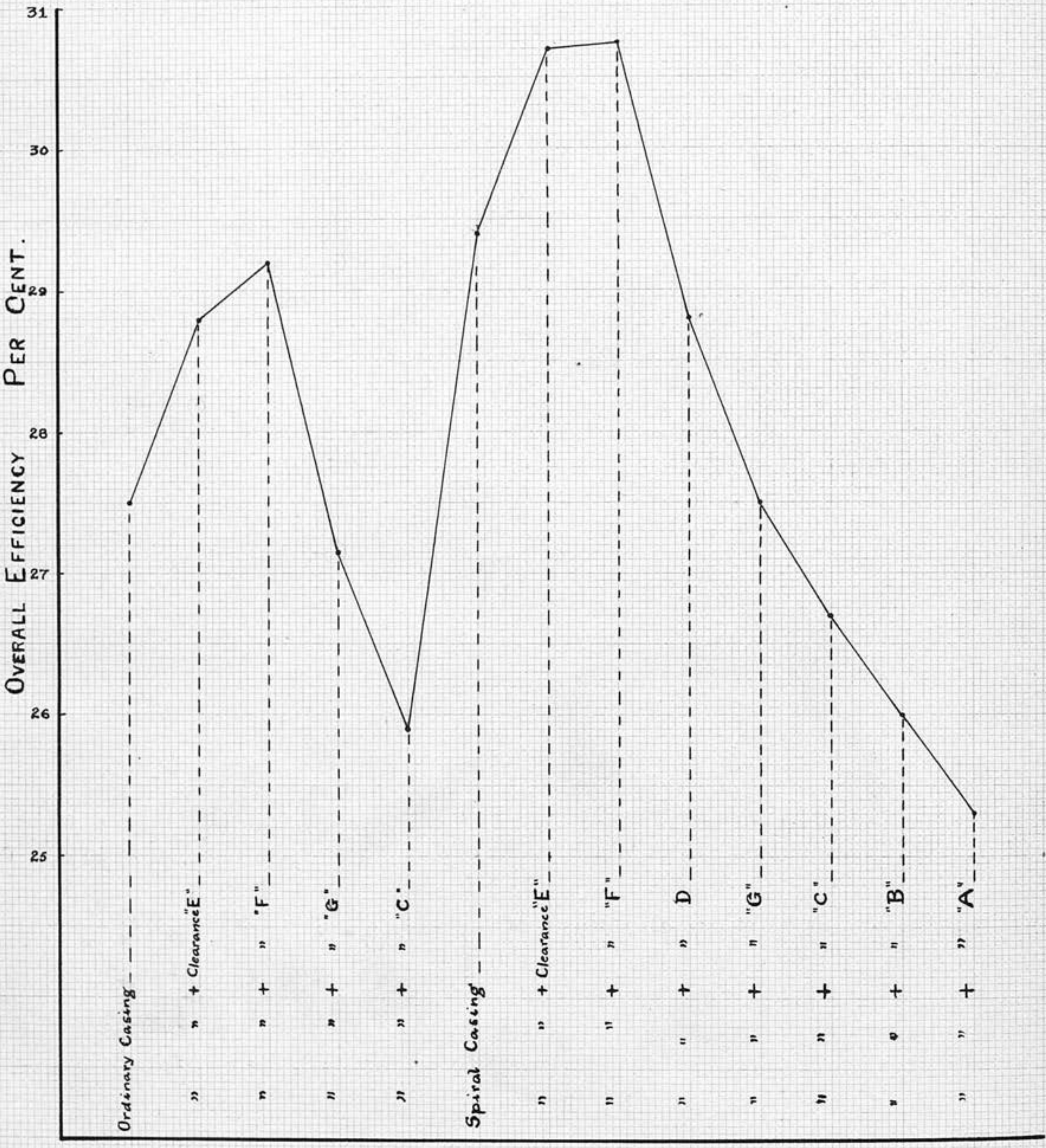




— Figure 76. —

The Various Clearances used in Experiments.

Figure 78. Illustrating Representative Results from Fan Casing Experiments with True Spiral and Variable Clearance at "Beak", at the same Fan Speed, 1100 revs. per minu[te]



efficiencies being plotted against a speed base.

Although the improvement effected by the substitution of the spiral fan for the ordinary fan casing is slight it is nevertheless appreciable.

No attempt was made to explore inside the improvised spiral casing.

(8) The Effect of Varying the Clearance between the Fan Runner and the Commencement of the Casing or "Beak" of the Fan.

From the investigation inside the fan casing it was clearly shown inter alia that about one-eighth of the volume dealt with by the fan re-entered the casing at the "beak". The clearance at this point was varied by the insertion of specially shaped pieces of wood, as illustrated in Figure 76. These were fitted, in turn, both to the improvised spiral and the ordinary casing, and exhaustive overall efficiency tests made.

As was expected several of the results obtained were but little different from each other and to show them all graphically would be confusing. Representative efficiency curves only are given in Figure 77; representative results for a constant fan speed are also set out in Figure 78. When the clearances made by using "A", "B", and "C" (Figure 76) respectively were employed, a sound resembling that of a high-speed motor was emitted when the fan was running; the pitch of the sound increased with the speed of the fan, and fell with increased clearance. Since the efficiency was lowered considerably with these clearances ("A" being the lowest) it was thus evident that the re-circulation had been checked at a greater expenditure of energy than occurred when the maximum clearance was allowed.

Compared with the overall efficiencies obtained with the ordinary casing alone, an average gain of 3.1 per cent. resulted from the use of the spiral casing and clearance /

clearance "E"; with the ordinary casing and clearance "F", the average gain was 1.55 per cent. These were the maximum average gains which resulted from this experimental work.

(9) Conclusions[1] (Section D).

(a) The variability in velocity of the air circulating in the fan casing is pronounced. This condition is chiefly due to:-

(i) the uneven distribution of the air through the fan;

(ii) the sudden increase of area into which the air enters immediately on emerging from the fan-wheel;

(iii) departures from the true spiral or volute in the manufacture of the fan casing.

(b) The velocity of air effluent from the runner of a Sirocco fan is considerably higher than the peripheral speed of such a fan; at some points of the discharge the air velocity may exceed the fan speed by over 70 per cent.

(c) The velocity of air re-entering the fan is higher than the peripheral speed of the fan. The general direction of the re-entrant flow coincides with that of normal flow.

(d) The effect of slight departures from the true spiral or volute is appreciable. Makers should endeavour to adhere more closely to the true curve.

(e) The transmutation of kinetic energy into pressure energy in fan casings of the type investigated, is most inefficient. The width of the casing in relation to the width of the fan must produce considerable loss of energy due to shock; it is analogous to a sudden enlargement where the head lost due to shock is $\left(\frac{V_1 - V_2}{2g}\right)^2$.

(f) Where /

1. The above conclusions are based solely upon the experimental work so far conducted, and are thus related to parallel-sided casings only, of a width considerably greater than that of the fan-runner, and where no diffuser arrangement is interposed between the fan-runner and the volute.

(f) Where no diffuser exists, it would seem preferable to design a casing to function solely as a collecting and guiding medium of the air discharged from the fan to the évasée, leaving the conversion of energy to the latter adjutage. (Note: A casing designed in accordance with this principle is being installed in the Mining Laboratory. See Appendix F).

(g) The shape of the casing at its commencement, i.e., at the "beak" of the fan, is of great importance. The marked throttling effect below the "beak" of a Sirocco fan, demonstrated in both Sections B and D, is caused by the reduction in section at that point. A casing should increase in section progressively from its commencement.

(h) An adjustable clearance between the fan runner and the commencement of the casing, such that the shape of the "beak" could be also adjusted to suit the clearance, would appear to be a distinct advantage.

Although not specially related to any of the foregoing conclusions the following may be added as a general conclusion:-

At the outset we endeavoured to emphasise the importance of effective and efficient ventilation in our deep and extensive mines, and especially in safety-lamp mines. Although our mining legislation makes no provision for a ventilating engineer at a colliery, the time seems opportune for the appointment of an official of such a character about a mine. While the manager must necessarily be capable of arranging a ventilation scheme which would be both practical and efficient, nevertheless his duties are so multifarious that he requires all the skilled assistance that can be given him. He is provided with a fully qualified surveyor, engineer and electrician /

electrician, together with their respective staffs, but in the most essential factor towards the successful exploitation of the mine, namely, ventilation, he has no such expert aid. The duties of the under-manager, oversmen and deputies are also diversified to such an extent that ventilation matters are usually considered only in emergency, although the last named officials have perforce to carry out their regular statutory inspections in this connection.

It would be quite impossible to assess the value of an efficient ventilating engineer at a mine. His sphere of activities would extend round the whole ventilating circuit. Underground, his regular and skilled inspection would lead to the removal of all sources of unnecessary and avoidable leakage, and to the efficient distribution of the air. His frequent consultations with the deputies could not fail to raise the standard of district ventilation to a high plane of health and safety. At the surface, the fan plant would be reliably and regularly tested and any glaring causes of inefficiency (e.g. natural ventilation) remedied as far as possible. If the fan was provided with an adjustable clearance, as has been suggested, its regulation for maximum efficiency would be assured; again, any structural alteration of the évasée which would lead to increased economy would be readily manifest to such an engineer.

In suggesting this appointment it is not intended that the ventilating engineer should be given authority to over-ride the manager in ventilation matters connected with the mine; the authority of the latter would remain, as now, supreme. Nevertheless, the criticisms and proposals made by such an engineer on all points pertaining to the ventilation of the mine would be invaluable to the manager. /

invaluable to the manager. The whole of the activities of this engineer would be concentrated on the fulfilment of Section 29 (1) of the Coal Mines Act (1911), and on assuring himself that the adequate amount of ventilation therein stipulated was being maintained with the maximum degree of efficiency possible. The measure of his success must inevitably be reflected in the balance-sheet.

PART V.

APPENDICES.

APPENDIX A:- To determine the Position within a Duct at which the Dynamic Tube will register the Mean Value of the Dynamic Pressure for the Section.

The following deduction is based upon the generally accepted assumption that the velocity distribution-curve of a section in a duct is elliptical in form, and therefore the position of the mathematically determined mean velocity exists at a point 0.127[1] of the diameter of the duct measured from the side. Many experimenters have investigated this position by actual observation; A. B. Eason tabulates nine results obtained by seven independent workers. The mean value of these results is 0.139 of the diameter measured from the side. There is thus a fairly close agreement in the theoretically determined position of the mean velocity and that obtained experimentally to justify the assumption. Eason also publishes results (page 162) which indicate that there is but little difference between square and round-sectioned ducts in this connection.

Since the static pressure is uniform over any given cross-section the variation in dynamic pressure is influenced only by changes in velocity over that section. Thus, if we determine the position in the duct where the mean value of v^2 exists for the section we shall have also ascertained where to fix the dynamic tube so as to measure the mean dynamic pressure at that section.

The outline ABNCD of Figure 1 is (to scale) the velocity distribution-curve of an air-duct of diameter AD (= 2a) in which the mean velocity is 175-feet per second - this figure was the mean velocity of the air passing through the rhone D (Fig. 48) during our experiments.[2] AB = DC = c = the velocity of the air close to the sides of the duct; the velocity at the centre = ON = b + c; BCN is a semi-ellipse.

The equation of that ellipse, with centre O, is:-

$$v = \frac{b}{a}\sqrt{a^2 - x^2} + c \qquad \text{.....................(1)}$$

The variable part of the dynamic pressure - the velocity head - is proportional to v^2. Figure 2 represents the velocity-head distribution where the abscissae are as in Figure 1, radii from the axis of the duct, but the ordinates, y, are proportional to the square of the ordinates in Figure 1.

The equation of B'N'C' is :-

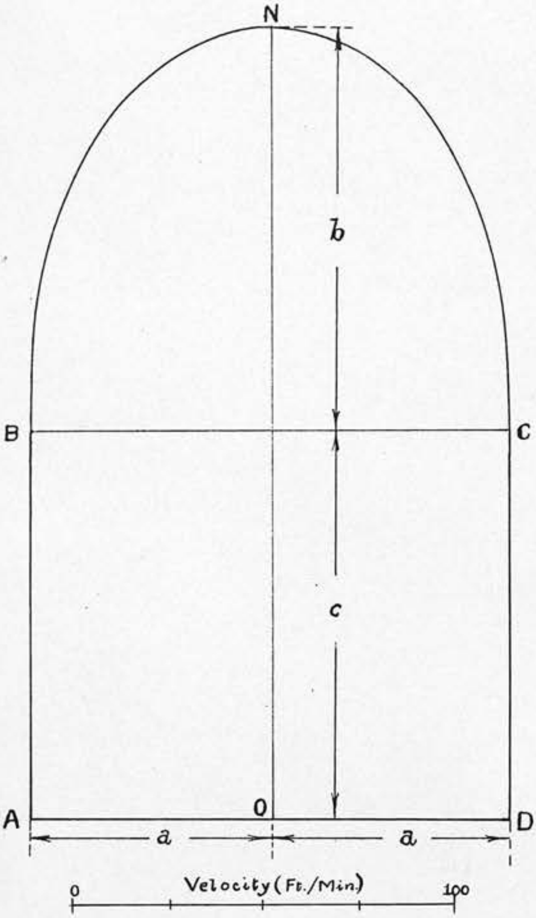
$$y = \frac{1}{2g}\left(\frac{b}{a}\sqrt{a^2 - x^2} + c\right)^2 \qquad \text{.....................(2)}$$

The mean ordinate of the solid figure made by revolving AB'N'C'D about ON' is:-

$\bar{y}$ //

---

1. "Flow and Measurement of Air and Gases" by A.B. Eason. (Griffith & Co., 1919) page 161.
2. Taking the values 5 or 9.4-feet per second (the extreme velocities employed in the fan-drift) or values 70 or 251-feet per second (the extreme velocities of flow in the wooden rhone) did not change the result.



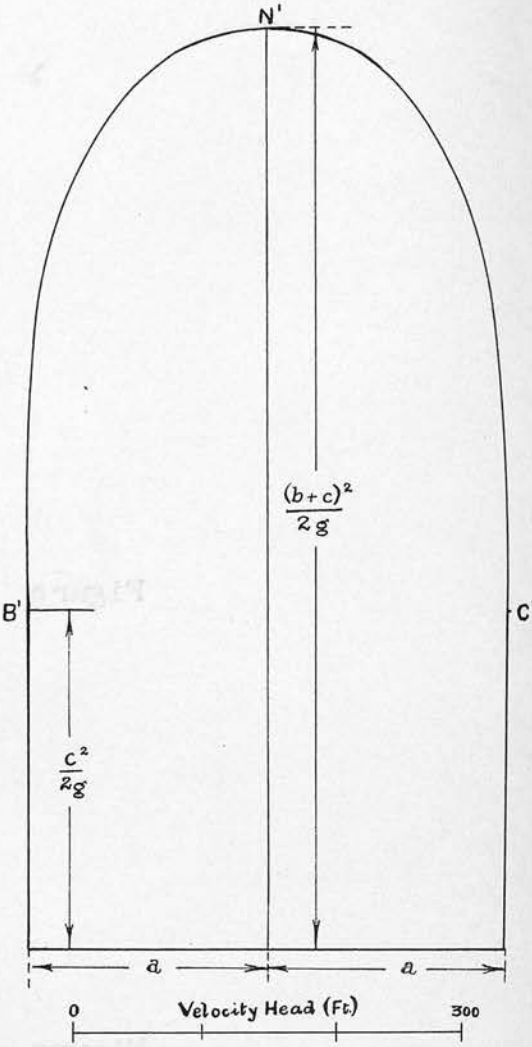


Figure 1. Velocity Distribution in Circular Duct.

Figure 2. Velocity-head Distribution in Circular Duct.

$\bar{y}$ = volume of figure of revolution $\div \pi a^2$

i.e. $\bar{y} = \frac{1}{\pi a^2}\int_0^a 2\pi xy.\,dx. = \frac{1}{a^2 g}\int_0^a x\left(\frac{b}{a}\sqrt{a^2-x^2}+c\right)^2.\,dx$

or $\bar{y} = \frac{1}{2g}\left(\frac{b^2}{2} + \frac{4bc}{3} + c^2\right)$....................(3)

To find the position of this mean ordinate, insert the value of $\bar{y}$ in equation (2) and solve for x; the result is:-

$$x = \frac{a}{b}\sqrt{\left\{b^2 - \left(c - \sqrt{\frac{3b^2 + 8bc + 6c^2}{6}}\right)\right\}} \quad \ldots.(4)$$

Here, we have four unknowns, and as we desire the value of x in terms of a only, the above result is useless unless we have the relationship existing between b and c. From the work of Messrs Stanton and Pannell[1] we learn that the value of mean velocity $\div$ velocity at centre varies little over the range of diameter and velocities we used, and that to adopt the value of 0.82 for this ratio over the whole series of our experiments will be sufficiently accurate.

The mean velocity in Figure 1 is:-

$$\bar{v} = \frac{2}{3}b + c$$

And since the velocity at the centre of the pipe is b + c, we have:-

$$\frac{\text{Mean Velocity}}{\text{Velocity at Centre}} = \frac{\frac{2}{3}b + c}{b + c} = 0.82$$

$$\text{From which,} \quad b = \frac{6}{5}c$$

Substituting this in equation (4), that equation reduces to:-

$\bar{x}$ = 0.725a.................................(5)

That is, the position for the mean dynamic gauge-reading will be 0.275 of the radius, or 0.137 (approx. one-seventh) of the diameter of the duct, measured from the side.

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1. "Similarity of Motion in Relation to the Surface Friction of Fluids", Phil. Trans., 1914, Vol. CCXIV, (A) page 199.

APPENDIX B:- Example of Calculation of Efficiency of a Diverging Duct.

The results tabulated hereunder are representative of the series of experiments carried out with divergent angles varying from approximately 2 to 34 degrees. They are concerned with a square-sectioned funnel of angle $7^o 10'$.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
	Dynamic water-gauge (in inches)					Velocity of air in feet per second	
Speed of fan Revolutions per minute	Rhone inlet (i)	Rhone Outlet (ii)	18 feet beyond évasée mouth (iii)	Loss of pressure in évasée (iv)	Quantity (q) in cubic ft. per second	Evasée inlet (v_1)	Evasée outlet (v_2)
600	0	0.020	0.775	0.755	30.11	120.44	13.38
700	0	0.025	1.055	1.030	37.29	149.16	16.57
800	0	0.036	1.388	1.352	44.15	176.60	19.60
900	0	0.052	1.801	1.749	50.65	202.60	22.50
1,000	0	0.052	2.288	2.236	57.86	231.44	25.70

(9)	(10)	(11)	(12)	(13)	(14)
	Energy lost through				
Kinetic energy in air at rhone inlet	Eddying turbulence, etc. in rhone	Eddying turbulence, etc. in évasée	Kinetic energy remaining at évasée mouth	Kinetic energy converted into pressure energy by évasée.	Efficiecy of évasée per cent.
Foot-pounds per second.					$100 \times \frac{a-(b+c+d)}{a-b}$
(a)	(b)	(c)	(d)	a-(b + c + d)	
507.5	3.2	118.2	6.27	379.8	75.30
964.5	5.0	199.7	11.90	747.9	77.94
1,601.0	8.35	310.3	19.72	1,262.6	79.27
2,417.0	13.7	460.6	29.81	1,912.9	79.60
3,599.0	15.6	672.8	44.41	2,866.2	79.98

Weight of 1 cubic foot air, under prevailing conditions of temperature, hygrometric state, and barometric pressure = 0.0748 pound.
Taking the observations when the fan speed was 900 revolutions per minute as an example, the figures in columns (9) to (14) were calculated as follows:-

Kinetic energy possessed by air at rhone inlet (a)

$$= \frac{qw \times v_1^2}{2g} = \frac{50.65 \times 0.075 \times 202.6^2}{64.4} = 2,417 \text{ foot-pounds per second}$$

Energy lost in rhone (b) = 5.2q (water-gauge ii) = 5.2 x 50.65 x 0.052 = 13.7 foot-pounds per second.

Energy lost in évasée (c) = 5.2q (water-gauge iv) = 5.2 x 50.65 x 1.749 = 400.6 foot-pounds per second.

Kinetic energy possessed by air leaving évasée (d) $= \frac{qwv^2}{2g} = \frac{50.65 \times .075 \times 22.5^2}{64.4} = 29.81$ ft.-lb./sec.

$$\text{Efficicinecy of évasée} = \frac{\text{Energy saved per second by évasée}}{\text{Total input of energy to évasée}} \quad \frac{a-(b+c+d)}{a-b}$$

$$= \frac{2,417 - (13.7 + 460.6 + 29.81)}{2417 - 13.7} = 0.796 \text{ or } 79.6 \text{ per cent.}$$

APPENDIX C:- Efficiencies of Divergence of an 8-Foot Duct with Various "Throat" Velocities.

(i) All four sides hading equally.
(ii) Two parallel sides, the other pair hading equally.

Angle of Divergence	Mean Throat Velocity (feet per second)		Efficiency of Divergence, per cent.		Remarks.
	(i)	(ii)	(i)	(ii)	
1°54'	110.5	86.8	51.7	10.4	
	131.3	105.7	52.4	13.8	
	152.0	122.7	53.2	14.4	
	171.4	137.7	53.0	13.9	
	188.4	151.7	52.5	13.1	
3°36'	124.5	106.7	71.8	45.0	
	145.6	126.9	71.0	45.5	
	172.4	148.4	72.1	46.1	
	193.6	170.7	72.1	47.1	
	212.8	185.3	72.0	45.3	
5°22'	130.6	117.9	77.5	54.7	
	150.5	139.5	77.0	55.4	
	175.3	157.3	77.0	54.3	
	197.3	175.3	77.2	53.9	
	216.8	193.3	76.7	53.3	
7°10'	120.2	122.8	75.3	65.6	
	149.2	142.3	77.9	65.5	
	176.6	163.7	79.3	65.8	
	202.8	186.7	79.6	65.8	
	231.4	206.7	80.0	63.2	
8°56'	106.9	120.7	74.2	68.0	
	133.4	144.0	75.6	69.2	
	149.2	165.3	75.9	69.5	
	170.1	188.0	76.1	69.1	
	186.9	206.3	75.3	68.7	
10°42'	92.5	121.3	64.3	69.5	
	112.1	142.8	66.4	70.5	
	128.1	163.9	65.1	70.5	
	144.3	181.3	65.7	69.5	
	165.0	201.5	68.5	69.4	
Elevation 12°)	82.8	117.9	49.0	68.6	
i))	104.4	137.2	57.1	68.6	
Plan 17°)	121.9	156.0	57.9	67.7	
ii) 13°52'	135.4	179.7	55.9(?)	68.8	
	154.0	200.0	57.4	68.5	
Elevation 24°)	70.2	111.0	31.2	64.0	(i) Duct
i))	84.0	128.1	34.8	63.5	4 ft.
Plan 34°)	102.4	151.2	41.2	64.6	long
	122.0	173.5	46.7	65.0	(ii) Duct
ii) 15°	135.2	185.7	46.8	63.4	5 ft.
					long.

Figures are averages of two separate tests.

In each of the five experiments in any series, the fan speeds were 600, 700, 800, 900 and 1000 revolutions per minute, and the gallery conditions were maintained the same throughout the complete series of tests.

APPENDIX D:- The following tables summarise the results of the detailed exploration made inside the 8-foot duct when diverging uniformly at $7^{\circ}10'$, such results being graphically represented in Figure 56. The explora: :tion covers five rates of flow, viz:-

(1) 30.1 cubic feet per second.
(2) 37.3 " " " "
(3) 44.2 " " " "
(4) 50.6 " " " "
(5) 57.9 " " " "

Table I. Variation in Dynamic Gauge-Readings per Foot of Length.

Distance from "Throat" (Feet)	Mean Dynamic Gauge-Readings. (Pounds per square Foot)				
	(1)	(2)	(3)	(4)	(5)
0	0.05	0.06	0.08	0.11	0.16
3	3.26	4.13	5.10	6.76	8.16
4	3.48	4.85	5.75	7.65	9.58
5	3.70	5.08	6.25	8.14	10.40
6	3.92	5.10	6.72	8.49	10.65
7	4.00	5.29	7.00	9.00	11.10
8	4.11	5.50	7.15	9.23	11.40

Note:- The figures given are the mean readings taken at each of the four possible theoretically correct positions for the dynamic tube to register the mean dynamic pressure at the particular section considered. (See Appendix A)

Table II. Variation in Efficiency per Foot of Length.

Distance from "Throat" (Feet)	Efficiencies of Divergence (Per Cent).				
	(1)	(2)	(3)	(4)	(5)
3	71.1	73.2	75.0	75.0	76.2
4	72.4	74.7	78.2	77.8	78.5
5	73.4	76.0	79.1	78.9	79.4
6	73.4	76.6	79.1	79.9	80.3
7	73.5	76.6	79.3	79.3	80.4
8	73.5	76.9	79.1	79.2	80.4

Note:- The above efficiencies were determined in the manner shown in Appendix (B).

APPENDIX E:- Divisional Velocities of Air at Seven Sections of the 18-inch Sirocco Fan Casing (See page 129)

Section No. 1 Area:- 10 inches x $2\frac{1}{4}$ inches.
Quantity passing:- 680.3 cubic feet per minute.

10"

3785	4645	4705	4880	4230	4082	o 4325	o 4295	o 4205	o 4660
3800	4315	4082	4800	4506	4435	x 4470	x 4405	o 4300	o 4210

$2\frac{1}{4}$"

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air ← Entering Fan.

Section No. 2 Area:- 10 inches x $2\frac{1}{2}$ inches.
Quantity passing:- 567.2 cubic feet per minute.

10"

o 3240	o 3590	3520	3712	3480	3070	2950	o 3919	o 4170	o 4250
o 4051	o 3740	3378	3400	3293	3790	o 3895	x 3850	o 3400	o 4127

$2\frac{1}{2}$"

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air ← Entering Fan.

x Indicates re-entrant velocity.

o Indicates marked turbulent conditions at particular division.

APPENDIX E (contd).

Section No. 4: Area:- 10 inches x 5 inches.
Quantity passing:- 1204.7 cubic feet per minute.

5"

2995	2865	2950	2848	2720	2740	2934	2950	3095	3400
3060	2240	2644	3880	3960	3455	3500	3600	3710	3785
2775	1820	2550	4350	4265	4465	3841	3630	o 3823	o 4042
o 2410	o 1090	4290	5275	5460	5625	o 4950	x 4550	x 4010	o 3946

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air
←
Entering Fan.

Section No. 5: Area:- 10 inches x 6 inches.
Quantity passing:- 1525.8 cubic feet per minute.

2958	3460	3640	3940	3985	3320	3060	3015	3868	2920
2900	3040	2825	3815	4450	4215	3690	3055	3554	3758
2720	2898	2400	4385	4635	4510	4130	x 3870	x 3730	o 4450
2500	2348	4778	5790	5350	5290	x 4370	x 4120	x 4015	o 4300

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air
←
Entering Fan.

x Indicates re-entrant velocity.

o Indicates marked turbulent conditions at particular division.

APPENDIX E (contd).

Section No. 6: Area:- 10 inches x $6\frac{1}{2}$ inches.
Quantity passing:- 1707.2 cubic feet per minute.

3860	3400	2865	3540	3980	3505	3540	3540	3570	3590
3721	3118	2800	3525	4150	3265	3320	3540	3688	3688
3980	3775	3465	4760	5350	3940	4055	x 4155	x 4300	o 4405
3815	3433	3900	5325	5270	5900	4102	x 4040	x 3775	o 4278

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air
←
Entering Fan.

Section No. 7: Area:- 10 inches x $7\frac{5}{8}$ inches.
Quantity passing:- 1872.8 cubic feet per minute.

3770	3475	3226	3812	4115	3170	3226	3365	3420	3750
3375	3505	3390	3935	4250	3390	3390	3440	3860	4165
4450	4298	3930	4605	4845	3840	3940	o 4000	o 4275	o 5105
o 3520	o 3700	4600	4970	4875	5105	4660	x 3940	x 4275	o 4900

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air
←
Entering Fan.

x Indicates re-entrant velocity.

o Indicates marked turbulent conditions at particular division.

APPENDIX E (contd).

Section No. 8: Area:- approx. 10 inches x 7 inches. (The projection of the radius through position No. 8 passes into the évasée - see Figure 65)

Quantity:-~~passing through the layers of section indicated above.~~ See below.

(4)	3145	2088	2479	3110	3110	3220	2355	2280	2355	2705
(3)	1788	2264	2747	2960	3141	2914	1972	2120	2570	2990
(2)	3005	2168	2450	3760	3630	2984	x 2740	x 2880	o 3226	o 3826
(1)	2180	1756	3422	4245	4400	3760	x 2845	x 2740	o 3302	o 3838

0 1" 2" 3" 4" 5" 6" 7" 8" 9" 10"

(Inches of Width)

← Relative Position of Fan. →

Direction of Air

←

Entering Fan.

Quantity passing through the layers of Casing Section indicated above.

(1) 224.2 Cubic Feet per minute (This quantity re-entered casing).

(2) 266.9 " " " "

(3) 419.2 " " " "

(4) 443.2 " " " "

x Indicates re-entrant velocity.

o Indicates marked turbulent conditions at particular division.

APPENDIX F. Determination of Size of Casing required of Width (1) 4-inches and (2) 6-inches for the 18-inch Sirocco Fan.

In Section A, Part IV, the effective width of the fan was seen to be confined to approximately the innermost 4-inches of its width. It is believed that a casing of exactly this width would effect an improvement in the efficiency of the plant. To determine the size of such a casing, or one of 6-inches width, the following procedure was adopted.

(A)

The maximum efficiency of the plant was obtained when the spiral casing and clearance "E" were used, the results then being:-

Fan Speed, 1250 r.p.m.; Quantity 2880 cub. ft. per min
Ventilating Pressure, 19.81 lbs. per sq. ft.;
Overall Efficiency, 34.4%

Taking the above quantity, the following table has been calculated, the assumptions being that:-

(i) the percentage discharge of air from the fan is as shown in Figure 27 (opposite page 87) when the fan speed was 1050 r.p.m.;

(ii) the fan is discharging all round its periphery over the width considered.

	Inches of Fan Width					
	1.	2.	3.	4.	5.	6.
Percentage Distribution	28.5	29.4	20.3	10.9	6.6	4.1
Percentage Volume, Cub. ft./min.	820	846	585	314	190	118
Radial Vel., (U), Ft/Sec.	34.8	36.0	24.9	13.3	8.1	5.0
Circum. Vel., (V) " "	163.5 feet per second					
Resultant Vel., (Vr)" "	146.5	149	132	116	109	104.8
$\frac{(Vr - V)}{(V)}$ 100 (Per Cent)	49.2	51.7	34.4	18.1	11.0	6.7

(1) The average resultant velocity over the four most effective inches of width is 135.9 feet per second.

The quantity (q) passing position No. 7 (See Figure 65) per second is,

$$\frac{2880}{60} = 48 \text{ cub. ft. per second.}$$

Cross-sectional /

APPENDIX F. (Contd).

Cross sectional area (a) required to pass this volume,

$$= \frac{48}{135.9} = 0.3535 \text{ sq. ft.}$$

Height of casing required at position No. 7

$$= \frac{3535 \times 144}{4} = \underline{12.7 \text{ inches}}.$$

(2) The average resultant velocity over the full width of the fan (i.e., 6 inches) is 126.2 ft. per second.

The required height at position Nol 7 is 9.12 inches.

From these respective heights, the spiral required for either a 4-inch width, or a 6-inch width of casing is readily determined.

(B)

Taking the quantity (2,442 cub. ft. per min.) when the fan speed was 1070 r.p.m., the ventilating pressure, 14.45 lbs. per sq. ft., and the efficiency 30.0%, and making the same assumptions as to distribution as in (A), the haight of the casing required at position No. 7 is exactly the same as calculated above, namely:-

For 4 inch width...................12.7 inches
For 6 inch width.................. 9.12inches.

As already mentioned, a casing 4-inches wide is being installed in the laboratory.

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